SAE Journal

Norman G. Shidle
Editor

Joseph Gilbert
Associate Editor

SAE JOURNAL PUBLICATION OFFICE
Business Press, Inc.
10 McGovern Ave.
Lancaster, Pa.

EDITORIAL OFFICE
29 West 39th St.
New York 18, N. Y.
Tel.: LOngacre 5-7174

ADVERTISING OFFICES
E. L. Carroll
Eastern Advertising Manager
29 West 39th St.
New York 18, N. Y.
Tel.: LOngacre 5-7170

A. J. Underwood
Western Advertising Manager
3-210 General Motors Bldg.
Detroit 2, Mich.
Tel.: TRinity 2-0606

•

SAE DETROIT OFFICE
808 New Center Bldg.
Tel.: TRinity 5-7495
R. C. Sackett, Staff Representative

SAE WEST COAST BRANCH
Petroleum Bldg.
714 W. Olympic Blvd.
Los Angeles 15, Calif.
Tel.: Prospect 6559
E. W. Rentz, Jr., West Coast Manager

TABLE OF CONTENTS

Bendix Products Division
CREATIVE ENGINEERING
GEARED TO QUANTITY PRODUCTION
75 MILLION BRAKES
Producing 75 million brakes is a remarkable accomplishment even in these days of large numbers and mass production methods. This remarkable record is made possible because Bendix Products Division has such things as —
A quarter century of specialized experience
Recognized engineering excellence
Research that has set the pace in design development
Proved manufacturing skill and capacity
Overwhelming endorsement of the automotive industry
Manufacturers who put their braking problems up to Bendix — regardless of the type of vehicle — soon see what this matchless experience in the fields of creative engineering and quantity production can do.
PRODUCED FOR THE AUTOMOTIVE INDUSTRY
BENDIX · PRODUCTS DIVISION · SOUTH BEND
Export Sales: Bendix International Division, 72 Fifth Ave., New York 11, N.Y. • Canadian Sales: Bendix-Eclipse of Canada, Ltd., Windsor, Ontario, Canada
Bendix
AVIATION CORPORATION
BUILDERS
OF THE BASICS
OF BETTER
MOTOR VEHICLES

Performance Prediction Method Proved Valid in Truck Tests

EXCERPTS FROM PAPER* BY

Carl Saal, Highway Transport Research Branch, Bureau of Public Roads

* Paper "Truck Road Performance—Actual Versus Computed," was presented at SAE Philadelphia Section, April 12, 1950. This paper will be printed in full in SAE Quarterly Transactions.

ROAD tests with seven trucks, tractor-semitrailers, full-trailer combinations to check the accuracy of a method[1] for predicting performance of commerial vehicles showed that:

1. The method is sound.
2. The basic factors now available for resistances to motion can be expected to produce average difference between computed and actual performance of 5 to 10% of computed performance.
3. The actual performance is always greater than the computed for operation on the level and is generally less than the computed for grades 3% and steeper.

[1] Expressions used in this method are given in the article. The method is detailed in SAE Quarterly Transactions, Vol. 4 (April), 1950, pp. 147–160, "Predicting Road Performance of Commercial Vehicles," by A. F. Stamm and E. P. Lamb, and in SAE Quarterly Transactions, Vol. 3 (April), 1949, pp. 215–228, "Evaluation of Factors Used to Compute Truck Performance," by Carl C. Saal. Both of these papers stem from work being done by the SAE Subcommittee on Classification and Evaluation of Transportation Engineering Formulas.

The seven test vehicles ranged from a two-axle, single-unit truck to a seven-axle tractor, semi-trailer, and full-trailer combination. Their grade-ability was observed in conjunction with a vehicle performance study conducted by the Committee on Economics of Motor Vehicle Size and Weight of the Highway Research Board. (Results of the grade-ability tests are not related to results of the parent study, which have been published as Research Report No. 9-A.)

Each of the seven vehicles was tested with three gross weights. Thus observations of actual grade-ability were obtained for 21 conditions of weight and power. Gross weights ranged from 20,000 lb for test vehicle No. 1 to 139,500 lb for test vehicle No. 7. Range in pounds per horsepower was from 177 to 553. Power developed by the engines varied from 112 to 263 net horsepower. All the vehicles were equipped with low stake and platform bodies.

Actual grade-ability was observed on 3, 6, and 8% grades and on an approximately level section of highway. Drivers were instructed to get the best

Table 1—Grade-Ability Calculation for Vehicle Operating with 50,000 lb Gross Weight, in Fifth Gear of Main Transmission and Overdrive of Auxiliary

Line No.	Items and computation	Engine speed in rpm 1200	1800	2400	2800
1	Theoretical road speed in mph $= \frac{\text{rpm}}{6.50^1 \times 8.37 \text{ (table 1)}}$	22.0	33.1	44.1	51.5
2	Net engine hp at sea level (from engine power curve)	87.0	128.0	154.0	160.0
3	Net engine hp at 2000-foot elevation = line 2 × .92	80.0	117.8	141.7	147.2
4	Rolling resistance hp = 50 × factor (from table)	28.1	46.7	68.0	84.0
5	Air resistance hp = 0.155 (area factor—table 3A) × velocity constant (table 3B)	4.1	14.1	33.2	53.0
6	Chassis friction hp (from table)	6.2	10.4	13.5	15.6
7	Level road hp = sum lines 3, 4, and 5	38.4	71.2	114.7	152.6
8	Reserve hp = line 2 − 6	41.6	46.6	27.0	..
9	Reserve hp per 1000 lbs = line 7/50	0.832	0.932	0.540	..
10	Grade ability in percent = line 8 × factor (from table	1.41	1.06	0.46	..

[1] Total gear reduction

possible performance from the vehicle when operating on a test section. A minimum of six runs was made over each section. Each test section was at an elevation of about 2000 ft above sea level.

Table 1 shows the step-by-step calculations for one combined gear ratio for test vehicle No. 4 operating with a 50,000-lb gross weight. Grade-ability is computed for four different engine speeds, ranging from the speed of peak torque to the speed of maximum horsepower.

It's interesting to note the portion that the individual resistances are of the level road horsepower.

Total resistance horsepower is 152.6 for an engine speed of 2800 rpm, or a road speed of 51.5 mph. Of this total, 55% is rolling resistance horsepower, 35% is air resistance horsepower, and only 10% is chassis

Method of Predicting Truck Road Performance

This method, like others before it, is based on the fundamental relation that the force available at the clutch (or its equivalent) is equal to the resistances that oppose motion of a vehicle. It differs from the others in that horsepower rather than torque is the common denominator; rolling resistance is varied with speed instead of being a constant; chassis resistance is expressed as a function of engine speed and size of the powered unit; and air resistance is introduced.

Basic performance equation, when horsepower available at a given elevation for doing work is balanced against the horsepower of the various resistances to motion, is:

$$NHP_e = RHP + AHP + FHP + GHP \qquad (a)$$

where:

NHP_e = net engine horsepower at given altitude

RHP = rolling resistance horsepower

AHP = air resistance horsepower

FHP = chassis friction resistance horsepower

GHP = grade resistance or acceleration horsepower (reserve)

The grade-ability is:

$$GHP = NHP_e - (RHP + AHP + FHP) \qquad (b)$$

When the grade resistance horsepower has been determined, the grade in percent may be found from this expression:

$$GHP = \frac{GW \times g \times mph}{37{,}500} \qquad (c)$$

$$\text{or } g = \frac{GHP/1000 \times 37.5}{mph} \qquad (d)$$

where:

GHP = grade resistance horsepower

GHP/1000 = grade resistance horsepower per 1000 lb of gross weight

g = grade in percent, feet of rise per 100 ft

mph = theoretical road speed, miles per hour

GW = gross vehicle or combination weight in pounds

The problem is to determine the reserve horsepower by formula (b), starting with the certified net engine horsepower at a given engine speed. The reserve horsepower is then converted into grade-ability by formula (d).

First step is to determine theoretical road speed which corresponds to the engine speed of any certified net engine horsepower. The following formula is used:

$$mph = \frac{rpm}{GR \times K} \qquad (e)$$

where:

rpm = engine speed, revolutions per minute

GR = combined gear ratio (product of axle and transmission ratios)

K = tire factor $= \frac{168}{r}$ in tire revolutions per minute at 1 mph

r = tire rolling radius in inches

Second step is to determine the net engine horsepower at a given altitude. The expression for this, for gasoline engines only, is:

$$NHP_e = (1 - 0.00004E)\ NHP \qquad (f)$$

where:

E = altitude, feet above sea level

NHP = certified net engine horsepower at sea level

After determining the net engine horsepower at a given altitude, the rolling resistance horsepower, the air resistance horsepower, and the chassis friction resistance horsepower must be calculated. Sum of the resistance horsepower is subtracted from the net engine horsepower at a given altitude to get the reserve horsepower. As long as the sum of the resistance horsepowers is equal to, or less than, the net horsepower available, the theoretical road speed can be sustained on the level.

The resistance horsepowers are calculated from the following formulas:

$$RHP = \frac{(7.6 + 0.09\ mph) \times mph \times GW}{375{,}000} \qquad (g)$$

$$AHP = \frac{0.0025\ A\ (mph)^3}{375} \qquad (h)$$

$$FHP = 1 + (0.0000002\ rpm \times GWR) \qquad (i)$$

where:

A = frontal area of vehicle in square feet

GWR = gross vehicle weight rating of powered unit in pounds

Obviously these formulas are rather involved and solution for any calculated road speed would be very laborious. Stamm and Lamb (see footnote on p. 17) have reduced the formulas to tables and charts which fit any given set of conditions. The solution simply involves extraction of a value from the tables or charts and filling in a simple computation sheet.

ction horsepower. At the lower road speed of 22 ›h. the percentages are 73, 11, and 16, respectively. Rolling resistance is the most important single ctor, with air resistance becoming important at ıe higher road speeds. In the case of 2800 rpm ıgine speed, the level road horsepower is slightly eater than the net engine horsepower available at 2000-ft elevation. For this reason there is no re-rve horsepower. This indicates that the theoreti-al road speed of 51.5 mph cannot be sustained on ıe level.

A calculation for each combined gear ratio was made in the same way shown in Table 1. Since test vehicle No. 4 had a five-speed transmission and a three-speed auxiliary, 15 such calculations were prepared for the 50,000-lb gross weight. The grade-ability curve for each combined gear ratio with grade-abilities of 9% or less is shown in Fig. 1. The curve marked "5th-0" was plotted with the grade-abilities computed in Table 1.

Maximum level of computed performance, which assumes that the driver shifts gears at the proper point, is shown in Fig. 2. Only the upper portion of the grade-ability curves shown in Fig. 1 for each combined gear ratio, is used in developing the maximum level of performance. A curve similar to the one in Fig. 2 has been prepared for each of the 21 gross weights, three for each test vehicle.

Actual Versus Computed

Actual sustained speeds observed on the 0, 3, 6, and 8% grades are compared with the maximum level of computed performance in Figs. 3 through 9. A chart has been prepared for the heaviest gross weight of each test vehicle. The sustained speed shown for the level section is the average of the speeds recorded for six test runs, three in each direction. Maximum speed sustained on any of six test runs is plotted for the 3, 6, and 8% grades.

All the test vehicles—except Nos. 1 and 3—have a five-speed transmission and a three-speed auxiliary transmission. Test vehicle No. 1 had a five-speed transmission and No. 3 had a five-speed transmission with a two-speed rear axle. A level road speed is not shown in Fig. 1 for test vehicle No. 1 since the engine was operating against the governor.

Except for the level section, there appears to be fairly close agreement between the actual and the computed results in terms of the speed in miles per hour. Actual ability for the three grades is less than computed in practically all instances. The opposite is true for the level section . . . in each case the actual is greater than the computed. For vehicle Nos. 2, 6, and 7 the difference is sizable.

Graphic comparisons between actual and computed grade-ability have been limited to the heaviest weight of each test vehicle. However, results for the other gross weights have been compared in the same way.

Table 2 summarizes the variation of actual from computed performance. This table includes the results for all the gross weights. Variation of the actual from the computed is expressed as a percentage; and a minus sign means that the actual was less than the computed.

Results in Table 2 indicate that the average variation, on a percentage basis, is about the same for

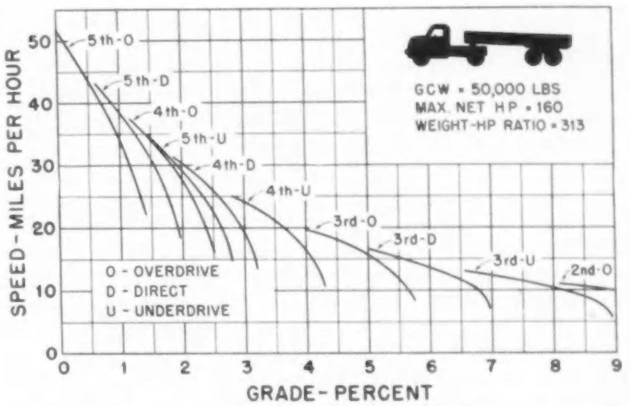


Fig. 1—Performance computed for each final gear reduction of test vehicle No. 4, which had a five-speed transmission and a three-speed auxiliary

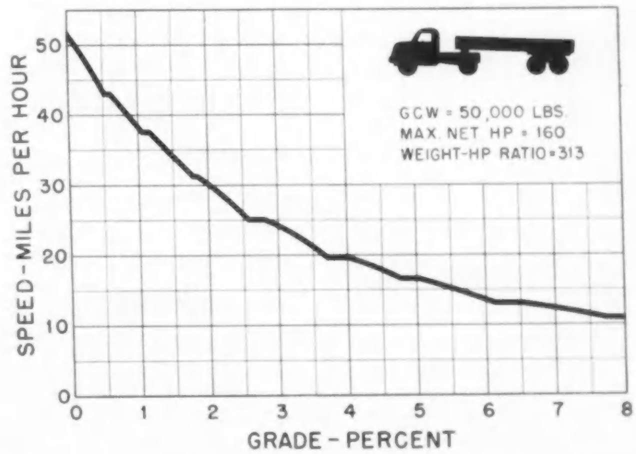


Fig. 2—Maximum level of computed performance for test vehicle No. 4, assuming the driver shifts gears at proper engine speeds

Table 2—Summary of Variation of Actual Performance from Computed Performance

Test Vehicle Number	Gross weight in pounds	Percentage of variation of actual from computed sustained speeds for grades of— Level	3 Percent	6 Percent	8 Percent
1	20,000	[1]	− 4.6	− 6.8	− 5.8
	24,000	[1]	− 4.6	+ 2.0	− 8.2
	26,500	[1]	− 2.9	− 2.0	− 7.4
2	36,500	+11.8	+ 7.2	+13.0	+ 4.1
	44,500	+10.0	+ 2.4	−10.3	0.0
	54,000	+15.3	+ 8.1	+ 4.8	− 1.3
3	33,000	0.0	−11.7	+ 6.2	− 8.5
	41,500	+ 1.1	− 9.8	+17.5	0.0
	50,000	+ 5.2	− 6.3	−15.7	−17.3
4	50,000	[1]	− 6.7	−12.5	− 7.2
	62,000	+ 2.1	− 7.2	−12.2	− 2.3
	74,500	+ 9.0	− 3.7	+ 2.1	−13.3
5	63,000	0.0	− 4.5	− 5.1	−13.3
	80,000	+ 6.8	− 8.7	−11.2	−11.0
	96,000	+ 8.8	− 5.3	−17.3	0.0
6	80,000	+ 5.0	+ 7.7	−12.2	−10.6
	100,500	+18.5	−15.0	+11.0	−20.0
	119,500	+23.5	+ 1.5	0.0	−20.0
7	93,500	0.0	−10.3	+ 7.8	−15.0
	116,500	+12.0	− 5.8	−11.0	− 6.2
	139,500	+19.5	−15.3	−11.7	−11.8
Average variation		8.7	7.1	9.2	8.7

[1] Engine operating against governor.

each of the test sections. Average variation ranges from 7.1 for the 3% grade to 9.2 for the 6% grade. Average percentage variation is equivalent to an average speed difference of about 4.0 mph on the level; 1.4 mph on the 3% grade; 1.0 mph on the 6% grade; and 0.8 mph on the 8% grade.

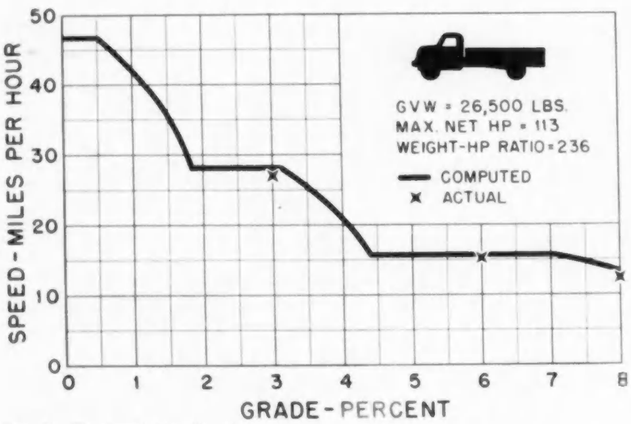


Fig. 3—Test vehicle No. 1

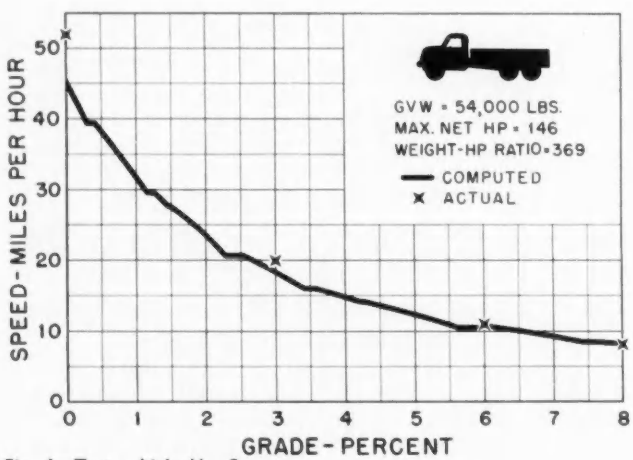


Fig. 4—Test vehicle No. 2

It's significant that the variation is always plus for the level and usually minus for the three grades. Since air resistance becomes an important factor at the higher speeds, the first thought is that the aerodynamic coefficient of 0.0025 might be too large. To illustrate how the resistance factors would have to be reduced to have the computed ability on the level equal the actual ability, the performance of test vehicle No. 2 with its heaviest weight will be considered.

Actual performance ability on the level in fifth overdrive was 52 mph as compared with computed ability of 45 mph. Net horsepower available at the specified altitude was 127; combined resistance horsepower (level road) was computed to be 162. This means that the resistance factors would have to be reduced by 35 hp in the required level road horsepower. Level road horsepower must equal the net available horsepower of 127, if the 52-mph road speed is to be sustained.

Computed level road horsepower is composed of: rolling resistance horsepower—92; air resistance horsepower—47; and chassis friction horsepower—23; for a total of 162 hp.

Reducing the aerodynamic coefficient from 0.0025 to 0.0018, the lowest value believed to be practical, would reduce air resistance horsepower from 47 to 34—a net reduction of 13 hp. Evidently the air resistance factor alone is not the reason for the variation between actual and computed performance.

There still remains a difference of 22 hp to be accounted for in the rolling resistance and chassis friction factors. This assumes that the coefficient of 0.0018 is correct. It is impossible to say which factors should be reduced and by how much without more reliable data.

Chief tools used in this method are the three resistance factors. These relations are based upon the best data available. Although they are recog-

Comparisons of Actual Performance of the

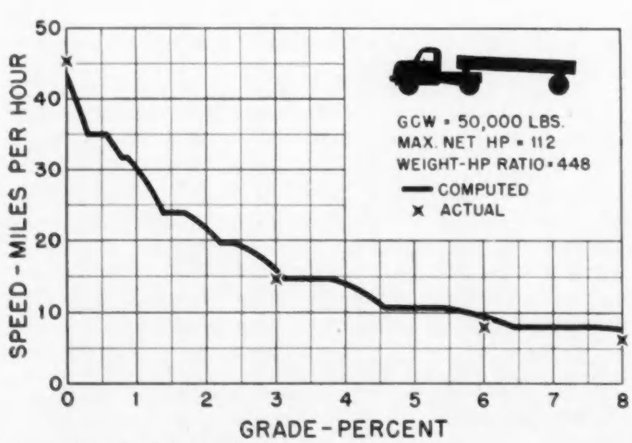


Fig. 5—Test vehicle No. 3

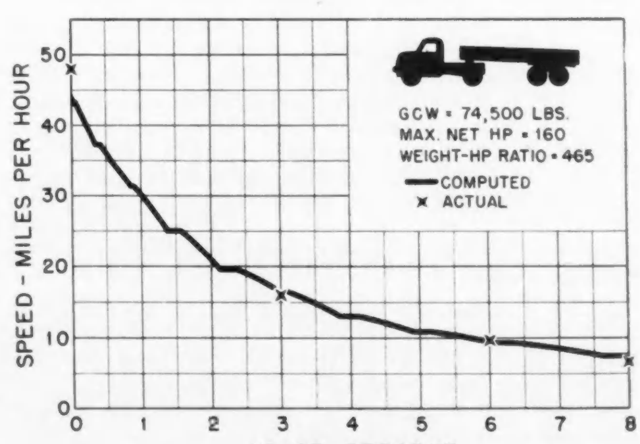


Fig. 6—Test vehicle No. 4

zed as not being the last word, this in no way compromises the method.

As more reliable data become available, it is only necessary to change the basic factors which are used in the method. But even when more reliable factors are available, the method cannot be expected to provide results that exactly match the actual performance of a specified vehicle. If exact ability is desired, actual road tests will continue to be the answer.

Chief deficiency, considering rolling resistance, is the almost total lack of basic data for commercial vehicles on surface types other than concrete. The rolling resistance increments deducted from the grade-ability for class 1 pavements are based on results of tests on passenger cars and light trucks. There is much work yet to be done for the lower type pavements and surfaces.

The data used in deriving the relation for rolling resistance on good class 1 pavements were obtained for two-axle single-unit trucks and three-axle tractor-semitrailer combinations, ranging in weight from 10,000 to 30,000 lb, and for road speeds of from 0 to 40 mph. The data have been extrapolated for speeds up to 80 mph and are intended for use with gross weights up to 100,000 lb. There is need for additional information, even on class 1 roads.

The power required to overcome air resistance becomes an important factor at the higher operating speeds. There is general agreement that the basic formula is valid; but there is doubt concerning the 0.0025 aerodynamic coefficient chosen by "expert" opinion.

It is hoped that eventually data will be available for determining more reliable coefficients for various classes of vehicles. Investigations now are being made of the possibility of full-scale wind tunnel tests and of a method that would use scale models and water as a medium instead of air.

Supporting data for chassis friction resistance are the least reliable. Although values were used for gross weight ratings of powered units ranging from 10,000 to 60,000 lb, basic data from which values were derived were observed for a 4 × 2 vehicles with ratings from 12,000 to 26,000 lb. The basic data were extrapolated considerably. Manufacturers have been asked to provide data during the last two years; but at this time only partial data on two vehicles have been obtained. The saving feature is that chassis friction is a relatively small portion of the combined resistance to motion.

It is not expected that average basic factors can ever be developed that will permit an exact prediction of the road performance of any specified vehicle. The varying characteristics of individual vehicles and the changing conditions of operation would preclude an exact prediction. Considering the serious gaps and uncertainties known to exist in the available factors, it is felt that factors eventually determined will produce potential performance within 5% of actual performance.

Paper on which this article is based is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.

Versus Computed Seven Test Vehicles

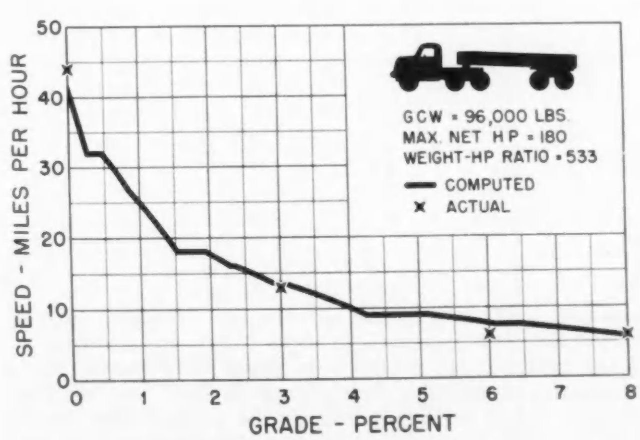


Fig. 7—Test vehicle No. 5

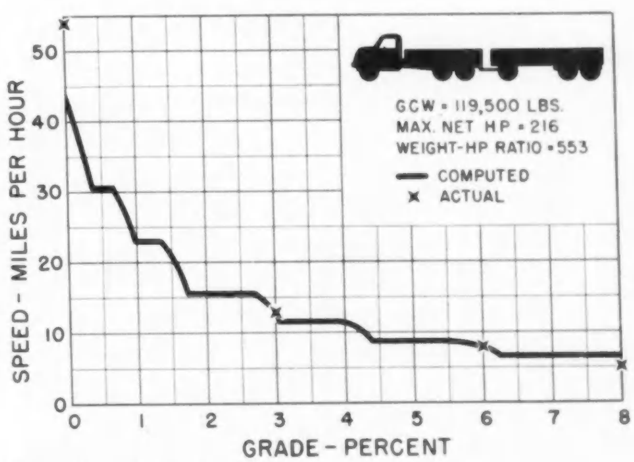


Fig. 8—Test vehicle No. 6

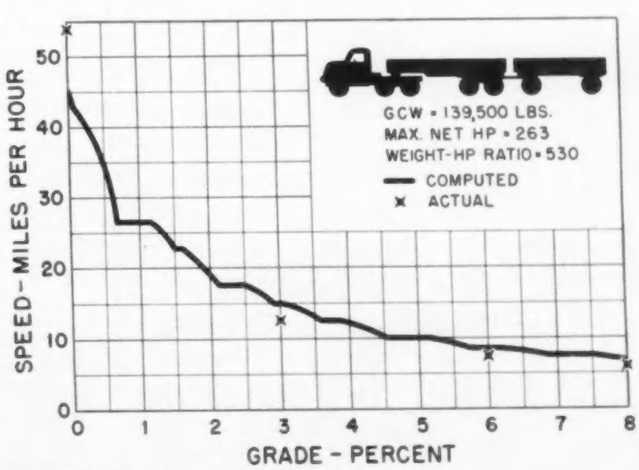


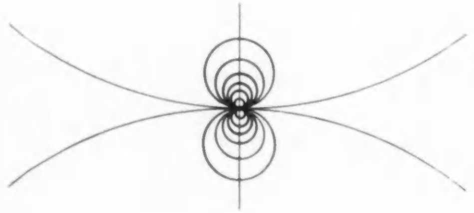
Fig. 9—Test vehicle No. 7

How CONTACT STRESSES

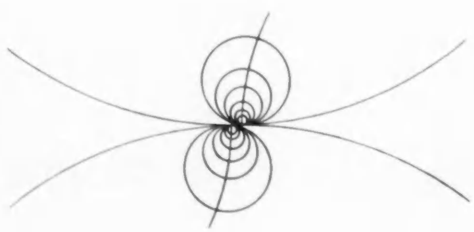
CONTACT stresses in gears subject outermost surface layers to reversing tangential stresses which lead to axial cracks. Contact stresses also result in a shear stress which reaches its maximum below the surface and can cause cracks running concentric with gear surfaces.

The stress patterns are revealed in photoelastic tests of cylinders and actual gear teeth. Cracks caused by contact stresses show up in photomicrographs of rolls run together with combined rolling and sliding action to simulate interaction of gear teeth.

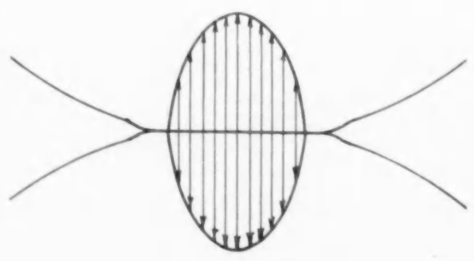
The stress patterns and their effects are explained below.



Two cylinders, under radial load only, develop a stress pattern in which the lines of constant stress are equal and opposite. This can be shown by stressing two transparent plastic cylinders and examining them by polarized light, which makes visible the stress pattern.



When torque is combined with radial load, the lines of constant stress are no longer opposite. They seem to start at either end of the elastically flattened surface of contact between the two cylinders. Intensities of the stresses under such conditions of combined loading are slightly greater than under radial load alone.



Deformation and compressive stresses are greatest at the center of the flattened area. At its edges, the material bulges.

When the cylinders are rotated, the bulges form elastic waves traveling ahead and behind the surfaces of contact. Surface material is subjected to tension ahead of the contact, then compressed, then tensed again. This reversed stressing tends to develop axial surface cracks.

Affect GEAR TEETH

BASED ON PAPER* BY

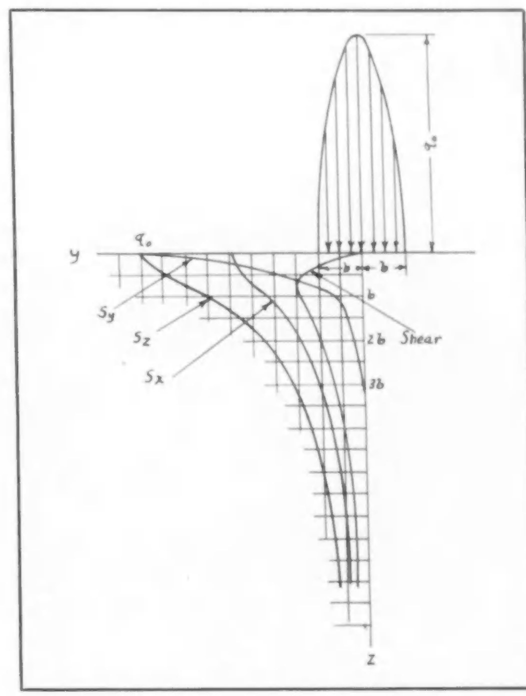
Earle Buckingham, Professor, Massachusetts Institute of Technology

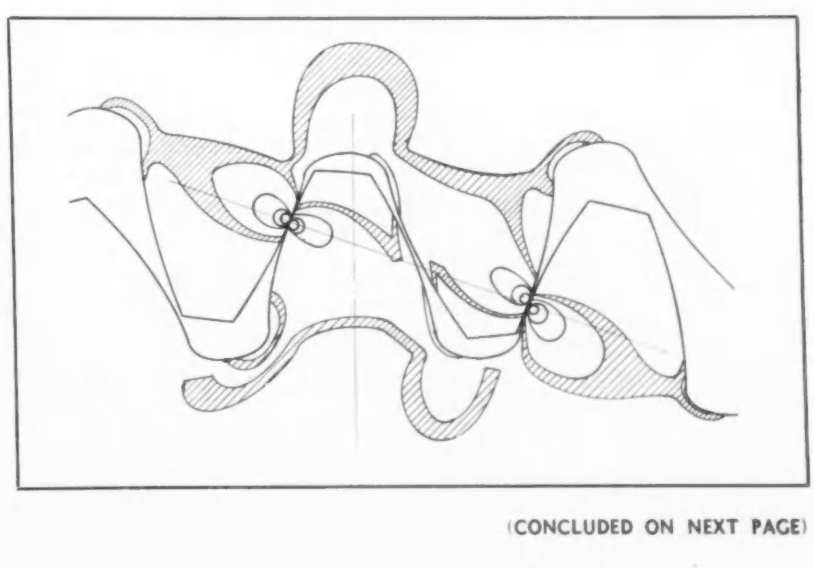
* Paper "Operational Stresses in Automotive Gears" was presented at SAE Summer Meeting, French Lick, Ind., June 7, 1950. This paper will be printed in full in SAE Quarterly Transactions.

The radial load applied by one cylinder to the other results in a comprehensive stress, S_z. There are also tensile stresses S_x in the axial direction and S_y tangent to the circle of the cylinder. All three are maximum at the surface, but their resultant shear stress reaches its maximum below the surface and is relatively high through an appreciable depth.

High subsurface shear stresses in gears tend to develop subsurface cracks roughly parallel to the surface of the gear teeth. Shifting of particles of cracked subsurface material leaves pits which destroy the usefulness of the surface.

If the depth of case on case-hardened gears is in the region of high shear stresses, cracks may start along the line between case and core. When the case is shallow, as with cyanide hardening and nitriding, the surface load capacity of the gears depends on the physical properties of the core material.





Drawing made from photoelastic tests of gear teeth shows combination of contact stresses with bending stresses and stress concentrations at fillets.

(CONCLUDED ON NEXT PAGE)

Photomicrographs confirm that large cracks do occur between case and core of case-hardened cylinders. Tests and service experience indicate that to develop full resistance to surface fatigue, the case should be twice as deep as the line of maximum shear stress.

Vertical surface cracks resulted from reversed stressing of the surface due to the traveling elastic wave. Wave opens cracks and admits lubricant. Then lubricant is trapped when contact flattens surface. Wave following contact frees lubricant—which lifts out tiny particles of the metal. Resulting shallow surface pits, often called incipient pitting, are seldom detrimental.

Combination of rolling and sliding induces plastic flow. Surface material work-hardens as it flows until it is stiff enough to resist further flow. Then rolling action picks up another portion of the surface material and carries it along until it, too, work-hardens enough to resist plastic flow and forms another corrugation.

On gear teeth of soft steel, particularly when the lubrication is inadequate, rolling combined with sliding tends to cause the surface material on the teeth of the driving gear to flow away from the pitch line, leaving a hollow, and causes the surface material on the teeth of the driven gear to flow towards the pitch line, leaving a ridge.

Paper on which this abridgement is based is available in full in multilithographed form from SAE Special Publications Department. Price 25¢ to members, 50¢ to nonmembers.

In the Next Issue

Editorial Coverage of two SAE National Meetings . . . . Tractor Meeting. . . . Aeronautic Meeting

Automotive Gear Steels....

How Industry Selects And Heat-Treats Them

EXCERPTS FROM PAPER BY*

V. E. Hense, H. H. Miller, and R. B. Schenck

Buick Motor Division, General Motors Corp.

* Paper "Economics of Automotive Gear Steels and Their Heat-Treatment," was presented at SAE Summer Meeting, French Lick, Ind., June 7, 1950.

PROPER selection of gear steel and correct processing of the material to produce desired properties and accuracy, including ease of fabrication, are musts for true gear economy. A survey of current practice followed by several automotive manufacturers shows what each has found to best satisfy his requirements for rear axle and transmission gears. As such, survey disclosures are assumed to represent economy in selection of steels and heat-treatments.

Factors in Selection

A steel should be selected because it fulfills all requirements of an application at the lowest cost. Proper choice can be made only when the engineer understands all conditions of manufacture. The requirements are dictated by the design, application, and quality level. Normal variation in the steel must be taken into consideration.

Cost of a gear is made up of the steel cost plus the cost of manufacturing. It is not possible in discussing economy to view steel cost without considering effects upon subsequent manufacturing operations.

Characteristics of a steel that influences economy are availability, cost, machinability, heat treating qualities, performance properties, and uniformity.

A steel must be available in sufficient quantities to fulfill production requirements. The limits of manufacture must satisfy the demands of the application. Critical requirements caused by design or quality demands may make it necessary to restrict the specification and thus increase the cost. If a particular specification has little use, it may be difficult to secure required quantities with suitable delivery.

There are many steels covering a wide range of properties available to choose from for automotive gear applications. The survey shows considerable variation among manufacturers in the selection of steel for each application. See Tables 1 and 2.

The base cost of the steel, in comparison with that of equivalent grades, is an important consideration in the selection. Any increased cost of one grade over another must be warranted by a necessary quality improvement or greater economy in processing.

Justification of a steel on the basis of decreased processing costs is often quite difficult because of intangibles involved. Items of cost in processing—such as tools, repairs, and scrap—can be evaluated only if adequate records are available for long production runs.

The steel selected must be capable of being machined to quality requirements at the specified production rate with a reasonable tool cost. Differences in inherent machinability between one specification and another should be considered before a choice is made. If available annealing cycles are inadequate for producing a structure with the required machinability, sufficient decrease in steel

Table 1—Automotive Rear Axle Gear Steels in Current Use (SAE Steel Designation)

Manufacturer	Drive Pinion	Ring Gear	Side Gear	Side Pinion
A	8617	8615	5130	5130
B	4620	4620	1019–1022	1022
C	4620H	4620H	8620H	8620H
D	8620	8620	8620	8620
E	4620	4620	1024	1024
F	1024	1024	1024	5120
G	4028	8620H	8620H	8620H
H	4028	4028	4032	4024
I	4620	4120 Mod.*	1137	1016

* Modification of the Former SAE 4120

Table 2—Automotive Transmission Gear Steels in Current Use (SAE Steel Designation)

Manufacturer	Counter Gear	Clutch Gear	Low Speed Gear	Second Speed Gear	Idler Gear
A	5145	5145	5145	5145	5145
B	4640H	4640H	4640H	4640H	4640H
C	8620	8620	8620	8620	8620
D	1320	1320 or 8620	1320	1320	1320
E	1024	1024	1024	1024	1024
F	5135H	5135H	5135H	5135H	5135H
G	4032[A]	4032[B]	4032	....	4032
H	1340H	1340H	1340H	1340H	1340H

[A] Counter Shaft Gears
[B] Drive Pinion

cost must be present to allow for the purchasing of necessary equipment.

Machinability is generally measured in terms of surface finish and tool life. On applications such as hypoid gears, where excellent tooth surfaces are necessary, finish is is controlling factor. Where surface smoothness is readily obtained, as in the shaving operations on transmission gears, tool life is the important consideration. In each case, however, the tool cost is a factor influencing economy.

Improvements in machinability that allow reductions in labor through greater speeds and feeds are most easily recognized. At equivalent production rates and quality, cost is affected only by tool life.

Sometimes improved machinability justifies the added cost of a sulfur addition. A sulfur addition may improve economy not only in its desirable effect upon finish, but may allow the production of greater tooth accuracy and less stress from machining. Sulfur additions which have proved beneficial to many other applications are not extensively used in steels for highly stressed gears.

The heat-treating characteristics of a steel must make it possible to get the specified properties throughout the range of chemistry and hardenability that must be accepted. If operational or processing demands are such that the full variation of a specification will not produce satisfactory heat-treating conditions, it is necessary to restrict the acceptable limits at an increased cost or to be able to divert extreme heats to other applications.

With some designs, difficulty may be experienced in securing satisfactory results at the minimum and maximum extremes of a specification. Difficulty with distortion causing erratic tooth changes and increased straightening costs are common at the high side of specifications; unsatisfactory response in securing the necessary properties may occur with low hardenability heats.

On jobs with critical requirements, it is often necessary to exert extra care and control in processing. Production pilot lots are used in addition to laboratory acceptance tests in determining if high and low hardenability heats of steel will produce satisfactory results.

Any narrowing of hardenability bands is a step toward more desirable production conditions and decreased manufacturing cost.

Usually there is little choice between steels of equivalent hardenability in their effect upon final operation of the gear. But where the design or application is such that the selection is made because of service characteristics, economy is affected. Such items as slight differences in wear resistance, toughness, or uniformity might become important considerations where the design and application are critical.

Steels that prove unsatisfactory generally fail in processing rather than in service requirements.

In the production of gears with great demands for interchangeability and a high production rate, uniformity is very important. Any improvement in uniformity of properties of one material over another cannot be overlooked as a factor influencing economy.

Heat-Treatment Considerations

Two general factors of cost must be considered in selecting a heat-treatment for automotive gears. The first is the economy of the operation itself, and the second is the effect upon subsequent processing.

Items affecting economy in the heat-treating operation are equipment, labor, operating, and maintenance costs. Additional factors are expense of idle equipment during down time and control cost.

Equipment is selected primarily to produce the required results at the specified production rate. The trend is to continuous, automatic equipment that furnishes uniform results at the minimum labor cost. Whether the equipment is heated by oil, gas, or electricity is a function of the cost of the fuel in the area of production. Choice of fixtures, refractories, and other maintenance items are somewhat regulated by the temperatures involved. The size of the equipment is determined by the production rate and the flexibility of operation desired.

Because the gear is changed during heat-treatment both in properties and dimensions, the effect upon subsequent processing must be considered. Changes in dimensions, referred to as distortion or movement, must be uniform in nature so that preventive corrections can be made in machining. Control of physical properties is very important in annealing cycles or in quench and temper cycles where the heat-treatment is followed by machining. Non-uniformity in heat-treating adds up to increased cost from straightening, repairs, or scrap.

Annealing is applied to automotive gears to produce a structure for satisfactory machinability and to relieve stresses which might cause distortion in subsequent operations. Machinability is measured in terms of finish and tool life. Continuous equipment is commonly used for annealing automotive gears. Factors influencing economy in annealing are labor cost, operating cost, maintenance cost, and the effect on cleaning operations. The method should be flexible, allowing changes to be made for steel variation.

In annealing rear axle gears, the development of a structure which will furnish the required smoothness of finish for tooth surfaces is the important consideration. Because of the expensive tooling and long setup times involved, machinability influences economy of processing to a great extent.

Annealing practice for rear axle gears, as indicated by the survey in Table 3, shows three general

Table 3—Annealing Practice For Rear Axle Gear Steels

Manufacturer	Drive Pinion		Ring Gear		Side Gear	
	SAE Steel	Treatment	SAE Steel	Treatment	SAE Steel	Treatment
A	8617	Normalize 1775 F	8615	Normalize 1775 F	5130	Normalize 1650 F
B	4620	Cycle Anneal 1750–1750–1175–1175–800–600 F	4620	Cycle Anneal 1750–1750–1175–1175–800–600 F	1019–1022	Normalize 1700 F
C	4620H	Cycle Anneal 1800–1150 F	4620H	Cycle Anneal 1800–1150 F	8620H	Cycle Anneal 1800–1150 F
D	8620	Cycle Anneal (1650–1700)–1220–1200–1180 F	8620	Cycle Anneal (1650–1700)–1220–1200–1180 F	8620	Cycle Anneal (1650–1700)–1220–1200–1180 F
E	4620	Cycle Anneal 1750–1825–1150–1000 F	4620	Cycle Anneal 1750–1825–1150–1000 F	1024	Bar Stock
F	1024	Normalize 1650 F	1024	Normalize 1650 F	1024	Normalize 1650 F
G	4028	Forging temperature to salt bath—1190 F for transformation	8620H	Forging temperature to salt bath—1190 F for transformation	8620H	Forging temperature to salt bath—1190 F for transformation
H	4028	Cycle Anneal at or above 1700 F for 2½ hr. Cool to 1100 F in 10 min.	4028	Cycle Anneal at or above 1700 F for 2½ hr. Cool to 1100 F in 10 min.	4032	Cycle Anneal at or above 1700 F for 2½ hr. Cool to 1100 F in 10 min.
I	4620H	Normalize 1700 F	4120 Mod.*	Normalize 1700 F Temper 1275 F	1137	Cycle Anneal 1500–1550–1130–1100–1075 F

* Modification of the Former SAE 4120

methods in use. The types shown are the cycle anneal, the normalize, and isothermal transformation from the forging operation.

From the cost standpoint, the cycle anneal and the normalize are about equivalent. A higher head temperature may increase maintenance cost. But generally slight variations in economy of operation are not important if improved results in machining or subsequent heat-treatment can be realized. A normalizing or cycle annealing furnace can be operated with the same labor cost.

An anneal, using the forging heat and transforming in salt at 119 F, offers a variation in processing in which there are possible savings. In this treatment it is possible to use the heat from the forging operation, to eliminate handling, and at the same time secure a structure which has good machinability. Another feature of this type of treatment is the descaling after annealing is made simpler in that a flash water quench from the transformation temperature loosens the scale. These savings must be weighed against the cost of operation of the salt bath.

Prime requisite in annealing transmission gears is to secure a structure which will satisfy the varied machining operations at an acceptable tool cost. The survey (Table 4) shows current annealing practice for transmission gear steels. Broaching, turning, milling, drilling, hobbing, and shaving operations are included in the manufacture of transmission gears.

It is sometimes necessary to produce a structure that will favor a broaching operation and consequently penalize turning operations. For instance, if finish requirements or design of a part makes the broaching operation critical, a harder structure is desirable. Tooth finish is readily controlled with shaving operations.

In hardening automotive gears it is desirable to produce the required physical properties with the least dimensional disturbance. Distortion can be caused by stresses from machining, non-uniform heating and carburizing, and variation in the hardenability of the steel.

Of the many factors involved in securing uniformly heat-treated parts, the quenching methods have probably been neglected the most. It is common practice to quench dense loads of gears in circulated oil at temperatures ranging from 100 to 130 F. With such practice, distortion may be held to close tolerances only with great difficulty.

Excessive distortion of gear teeth and shaft dimensions, along with non-uniform changes in ID dimensions, are major problems for the plant striving for a uniform end product. Where such distortion becomes unbearable, it is often economical to

Table 4—Annealing Practice For Transmission Gear Steels

Manufacturer	SAE Steel	Method	Temperature, °F
A	5145	Cycle Anneal	1650–1260
B	4640H	Cycle Anneal	1800–1200
C	8620	Cycle Anneal	1650–1700–1220–1200–1180
D	1320	Cycle Anneal	1750–1825–1150–1000
E[A]	1024	Normalize	1650
F[B]	5135H	Forging Temperature to Salt Bath	Transformation Temperature 1260
G	4032	Cycle Anneal	At or above 1700—2½ hr—cool to 1100 in 10 min.
H	1340H	Cycle Anneal	1475–1600–1130–1170

[A] Second Speed and Idler Gears Not Normalized
[B] Idler Gear Normalized 1850 F

incorporate a hot quench with either an oil or salt quenching medium. Such cycles, although minimizing distortion, do not eliminate it.

Since hot quenching treatments currently are generally more costly than the conventional quench, the improvement in uniformity must be balanced against the increased expense. Unless considerable scrap or rework is eliminated, it is often impossible to justify a hot quenching treatment. Die quenching is sometimes used for maintaining a dimensional accuracy during the quenching operation.

After hardening, straightening is sometimes necessary. This is not a desirable operation and the cost is high. Every effort should be made to eliminate straightening wherever possible.

The practice in the carburization of rear axle gears is quite uniform throughout the automotive industry. (See Table 5.) Gas carburizing is most popular, with some carbonitriding and pack carburizing being used. Case depth practice for ring gears and drive pinions is quite uniform among manufacturers with some variation in the treatment of side gears and side pinions.

Oil is used predominantly as the quenching medium. Rear axle ring gears are die quenched to maintain the flatness of the back face and the roundness of bore. Drive pinions are mass quenched from the carburizing operation by some manufacturers and singly quenched by others. Buick is installing die quenching machines for drive pinions. No hot quenching was reported in the survey for hardening rear axle gears.

The practice in tempering rear axle gears varies, with some manufacturers reporting no tempering operation. One manufacturer does not temper ring gears. He removes the part from the quenching press while it is hot enough to produce a tempering effect.

Table 6 gives survey results on heat-treatment of transmission gears. In the manufacture of automatic transmission gears, it has been necessary on some applications to go to great lengths in the

Table 5—Heat-Treatment For Rear Axle Gear Steels

Manufacturer	Part Name	SAE Steel	Method	Case Depth (Inches)	Quench Medium	Tempering Temp., °F
A	Drive Pinion	8617	Pack Carburize	0.045–0.060	Oil	325
	Ring Gear	8615	Pack Carburize	0.040–0.055	Oil	325
	Side Gear	5130	Carbonitride	0.005–0.015	Oil	325
	Side Pinion	5130	Carbonitride	0.005–0.015	Oil	325
B	Drive Pinion	4620	Gas Carburize	0.050–0.055	Oil	300
	Ring Gear	4620	Gas Carburize	0.050–0.055	Oil	300
	Side Gear	1019	Pack Carburize	0.050–0.055	Caustic	350
	Side Pinion	1022	Pack Carburize	0.050–0.055	Caustic	350
C	Drive Pinion	4620H	Gas Carburize	0.043–0.048	Oil	None
	Ring Gear	4620H	Gas Carburize	0.043–0.048	Oil	None
	Side Gear	8620H	Gas Carburize	0.043–0.048	Oil	None
	Side Pinion	8620H	Gas Carburize	0.043–0.048	Oil	None
D	Drive Pinion	8620	Gas Carburize	0.040–0.050	Oil	None
	Ring Gear	8620	Gas Carburize	0.040–0.050	Oil	None
	Side Gear	8620	Gas Carburize	0.025–0.035	Oil	None
	Side Pinion	8620	Gas Carburize	0.025–0.035	Oil	None
E	Drive Pinion	4620	Gas Carburize	0.040–0.050	Oil	None
	Ring Gear	4620	Gas Carburize	0.040–0.050	Oil	None
	Side Gear	1024	Gas Carburize	0.040–0.050	Oil	300
	Side Pinion	1024	Gas Carburize	0.040–0.050	Oil	300
F	Drive Pinion	1024	Gas Carburize	0.030–0.040	Oil	350
	Ring Gear	1024	Gas Carburize	0.030–0.040	Oil	None
	Side Gear	1024	Gas Carburize	0.025–0.030	Oil	350
	Side Pinion	5120	Gas Carburize	0.025–0.030	Oil	350
G	Drive Pinion	4028	Gas Carburize	0.045–0.055	Oil	375
	Ring Gear	8620H	Gas Carburize	0.045–0.055	Oil	375
	Side Gear	8620H	Gas Carburize	0.030–0.040	Oil	375
	Side Pinion	8620H	Gas Carburize	0.030–0.040	Oil	375
H	Drive Pinion	4028	Gas Carburize	0.040–0.050	Oil	None
	Ring Gear	4028	Gas Carburize	0.040–0.050	Oil	350
	Side Gear	4032	Gas Carburize	0.030–0.040	Oil	350
	Side Pinion	4024	Gas Carburize	0.030–0.040	Oil	350
I	Drive Pinion	4620H	Pack Carburize	0.040–0.050	Oil	None
	Ring Gear	4120*	Pack Carburize	0.040–0.050	Oil	325
	Side Gear	1137	Carbonitride	0.015–0.025	Oil	None
	Side Pinion	1016	Pack Carburize	0.040–0.050	Water	None

* Modification of the Former SAE 4120 Specification

selection of heat-treatments to provide the necessary accuracy and uniformity. (See Table 7 for survey findings.) Selective hardening of gear teeth and hot quenching treatments are being used extensively. Design has influenced the application of heat-treatments to a great extent. Some designs demand selective hardening, not only for maintaining tooth accuracy, but also because parts could be furnace hardened only with great difficulty.

Hot quenching operations are being carried out with either salt or oil as the quenching medium. Some hot quenching is being done in continuous furnaces where the quenching unit is conveyorized. In other applications the parts are heated in continuous furnaces and batch quenched.

Probably the most significant move in the heat-treatment of transmission gears has been the development of continuous gas carburizing or carbonitriding equipment. This equipment has been applied quite generally over the last 15 years and has probably been the greatest single contributing factor to the uniformity of heat treatment of transmission gears. Uniformity of heating and quenching, flexibility in controlling case compositions, and high production are typical advantages of this heat treating equipment.

Continuous heat treating usually proves most economical from the labor standpoint. In selective hardening or batch type hot quenching, the parts heat treated per man-hour are generally decreased considerably. For example, on one job a part was changed from a continuous furnace operation to a rotary furnace with a batch quench in molten salt; the gross production decreased from 370 parts to 159 parts per man-hour. This particular move was made to reduce rework and improve quality, but the labor cost was more than doubled.

In the selective hardening of gears, production rates of slightly over 100 pieces per man-hour are typical. If such parts could be hardened continuously, the production rate could be greatly increased with a corresponding labor saving. Therefore, if possible, parts should be processed in continuous equipment.

Paper on which this article is based is available from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.

Table 6—Heat Treatment For Transmission Gear Steels

Manufacturer	SAE Steel	Method	Case Depth (Inches)	Quench Medium	Tempering Temp., °F
A	5145H	Cyanide	....	Oil	480
B	4640H	Gas Carburize	0.008-0.012	Oil	425
C	8620	Gas Carburize	0.040-0.050	Oil	325-340[A]
D	1320	Gas Carburize	0.040-0.050	Oil	350
E	1024	Gas Carburize	0.025-0.030 0.030-0.040[B]	Oil	350
F	5135H	Gas Carburize	0.008-0.012 0.012-0.016[C]	Oil	360
G	4032	Gas and Pack Carburize	0.030-0.040	Oil	400[D] 480[E] 600-620[F]
H	1340H	Carbonitride	0.005-0.010	Oil	450

[A]—Some Hot Quenched In Oil
[B]—Low And Second Speed Gears
[C]—Clutch Gear
[D]—Drive Pinion Gears
[E]—Countershaft And Low Speed Gears
[F]—Reverse Idler Gear

Table 7—Automatic And Semi-automatic Transmission Gear Practice

Manufacturer	SAE Steel	Annealing Treatment	Hardening Method	Case Depth (Inches)	Quench Medium	Tempering Temp., °F
A	5145	....	Carbonitride	0.007-0.010	Oil	440
	1141	....	Induction	....	....	700
B	5140H	Cycle Anneal (1600-1625)-1325-1200-1100 F	Gas Carburize	0.004-0.007	Oil	325
	5140H	Cycle Anneal (1600-1625)-1325-1200-1100 F	Neutral Atmosphere	....	Oil	500
	5145H	Bar Stock	Gas Carburize	0.005-0.010	Oil	500
C	1024	....	Gas Carburize	0.025-0.030	Oil	350
	8620H	Normalize 1650 F Temper 1250 F	Gas Carburize	0.025-0.030	Oil	350
D	4032	Cycle Anneal at or above 1700 F for 2½ Hr. Cool to 1100 F in 10 Min.	Gas Carburize	0.030-0.040	Oil	400
E	1052	Quench 1550 F Oil, 1180-1220 F Temper	Induction	0.025-0.050 Below Root of Tooth	Oil	450
	1330H	Bar Stock	Gas Carburize	0.008-0.012	Salt	450

Farm Machines Go to Work

THE scarcity and high cost of hand labor in 1935 and 1936 forced Hawaiian planters to cultivate, fertilize, and harvest their sugar cane mechanically with machines, such as those described in this article. At present there are less than one-quarter the number of field workers than there were, employed on 250,000 acres of plantations scattered over five major islands.

There are two types of cultivation—unirrigated cane in wet areas and irrigated cane in dry areas, where cane is planted in the bottoms of "kuakuas" or furrows and spaced in rows 5 ft apart.

In the irrigated cane land, discussed here, track-laying tractors hold a decided advantage. They can travel on the tops of kuakuas without sliding off or unduly breaking them.

Some of the implements shown may seem crude. But most of them were built in open-air welding shops without benefit of designs, drawings, or technically-trained personnel. They are the product of dire need, a few ideas, some hot-rolled steel plate, and a welding torch. In some instances there are as many variations of a single type of tool as there are plantations. This is due to different conditions, to design evolution and lack of standardization.

Practically all the implements are Island-built, by the plantation, the tractor dealer, or one of the shops in Honolulu commissioned by the sugar factor.

1. Plowing: Track-type tractors were used for plowing and sub-soiling operations on Hawaiian plantations long before the beginning of the mechanization program. The mold-boar plow shown here is working in tough conditions. It is hydraulically operated, lifting the shares vertically out of the ground.

Most plowing operations involve more than turning over the soil. It must be stirred up and old cane stools broken up so they will not start a new growth of inferior cane.

2. Sub-Soiling Operations: In conjunction with plowing, most fields are subjected many times over to sub-soiling operations. Because underground rocks are often hit, the standards are pivoted and usually have some type of shear pin or spring release to prevent damage to the equipment.

In one type sub-soiler, two individually-pivoted standards, mounted on a parallelogram frame, penetrate to about a 24-in. depth. This view shows the hydraulic mechanism, leading edge of standards, and V-type bottoms.

n Hawaii's Sugar Fields

EXCERPTS FROM PAPER* BY

Jay D. Johnson, Caterpillar Tractor Co.

* Paper "Mechanization of the Sugar Industry in Hawaii," was presented at SAE Central Illinois Section, Peoria, Feb. 27, 1950

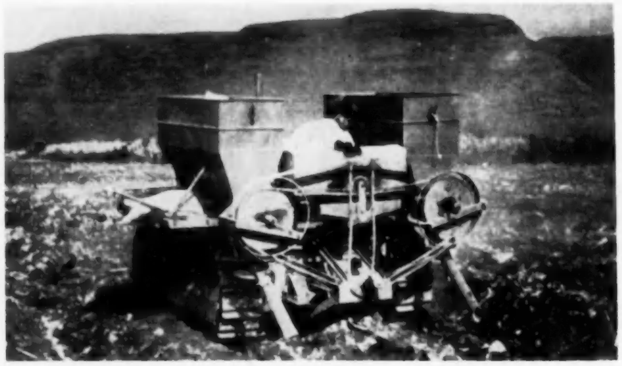
3. Furrowing: After plowing and sub-soiling operations, the kuakuas are formed with furrowing plows. These are really oversize middle-buster bottoms. Many fields are laid out with surveyor's transit to get the right slope and length of lines for best irrigation.

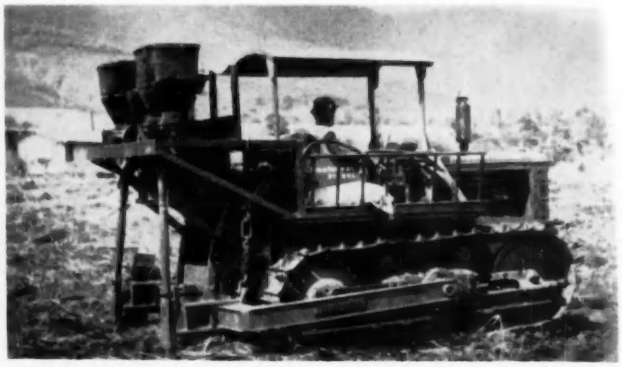
4. Seeding: To replace hand planting of seed cane, plantations have cobbled up many so-called planters that carry as much seed as the outrigger platform will hold. This unit also is equipped with fertilizer bins that sprinkle a small amount of fertilizer with the cane.

5. Line Reshaping: In the sugar growing cycle, from three to four harvests of cane are taken from each planting of a field. The first harvest on "plant cane" is made after about 24 months. Subsequent harvests on "ratoon cane" are made at from 18 to 24-month intervals.

The kuakuas or furrows break down with the weather and almost flatten completely after a harvest. They must be built up again with a tool similar to the one shown at right.

This is a two-row line reshaper on ratoon cane. The photo shows the compound angle at which the discs are mounted, with their axes inward and slightly forward, so that movement of the tractor keeps them rotating slowly. The discs are commercial 32-in. plow discs, except that the cutting edges are cupped to pick up more soil.





6. Fertilizing: Fertilizing is an operation in which tractors have entirely replaced hand labor. Only exceptions are the later stages of growth or in wet areas where it is impossible to enter the field. Greatest difficulty with mechanical fertilizers is that they "gum up" in augers and agitators from high humidity and unexpected rainfall. This attachment is driven from the power take-off.

7. Knifing: This machine is a combination fertilizer and knifer. After the cane has been "ratooned" two or three times, the cane stools, or root systems of the cane plants, become so large that they send up too many new shoots for healthy growth.

After a harvest, these knifers make one pass down the side of the cane row to cut away a portion of the root system.

8. Harvesting: Most of the research and development by the plantations and the Hawaiian Sugar Planters' Association has been expended on cane harvesting. Due to both the long growth period and hybrid cane varieties, the yield averages better than 80 tons per acre throughout the territory. It is not unusual to see a field go to 130 or 140 tons on some plantations on Kauai or Oahu. By comparison, the yield in Louisiana and Florida is from 30 to 50 tons, and in the Caribbean, from 40 to 70 tons.

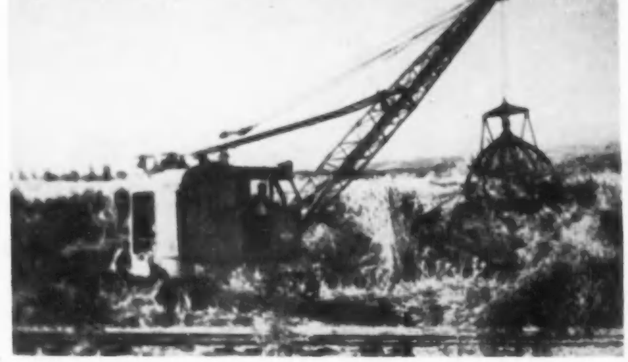
The cane does not always grow upright and intertwines itself until it is impossible for a worker to walk or even crawl through a field. The cane reaches a length of 20 to 30 ft in many instances, although much of this growth is horizontal.

The photo above, at left, shows a harvesting method that is being rapidly replaced. It is called "grab harvesting." The cane is pulled from its moorings by the crane and then loaded directly into rail cars or trucks.

Another method of harvesting, shown above at right, is to pull a "drag rake" back and forth between two units equipped with winches. The cane is piled into windrows, where it is later loaded with cranes, equipped with grabs, on trucks or cane cars.

All the irrigated cane and most of the unirrigated cane on the Islands is burned immediately prior to harvesting. Workers will set fire to the windward side of a field to burn off the cane trash. This eliminates having to strip it by hand or some mechanical means not yet perfected.

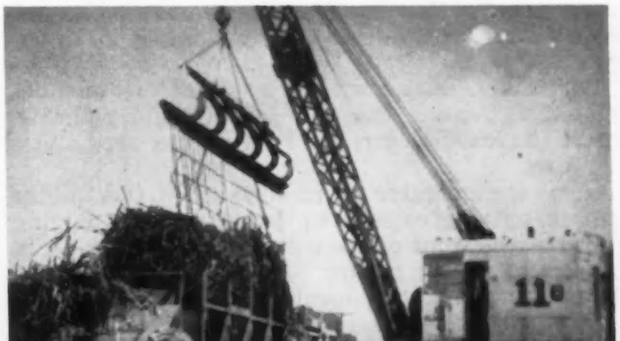
9. **Cutting:** Most cane cutting or harvesting is done with push-rakes mounted bulldozer-fashion on the front of tractors. Two large duckbill-shaped knives, extending out front of the rake, travel in the bottom of the kuakuas and shear off cane at, or slightly below, ground level.



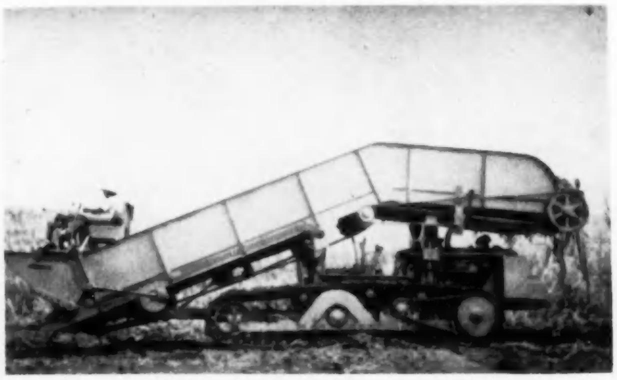
10. **Loading:** From the windrows the cane is loaded into rail cars or trucks for transportation to the mill. Because railroads, rolling stock, and locomotives are becoming obsolete, and replacement costs are high, there has been a scramble for better transportation methods.

11. **Transportation:** While large trucks are used for most motorized transport, the tractor-trailer method is becoming popular. Shown at left is a Caterpillar DW-10 dumping 15 tons of cane at a mill stockpile. It's important to get burned cane to the mill quickly because juice transformation from sucrose to glucose is rapid. Percentage of sugar goes down while that of molasses goes up faster than in unburned, hand-cut cane.

The Kenworth truck at right meets highway load and width regulations. But the cane projecting over the side does not.



12. **Unloading:** This LeTourneau unit carries about 25 tons of cane. The method of unloading is quite efficient. The chain sling on the inside of the stake body is fastened to the top of one side of the trailer. The other side of the sling hooks over the opposite side by means of a U-shaped steel channel. When the load reaches the mill, a crane with special hooks rolls the cane over the side of the retaining wall (right). The cane goes on a conveyor to the cleaning plant before crushing in the mill.



13. Harvester Development: Much research and development is being conducted on bigger and better harvesters. Aim is to bring cane into the mill cleaner and less mutilated than by present methods. Five or six type harvesters are being tested for various conditions.

The one shown at left, above, has a one row-cutter that was not too successful chiefly because of its inability to turn around in the field. The power unit is a Caterpillar D-6 tractor with extended track to the rear. Front of the unit, which is the rear of the tractor, can be raised and lowered hydraulically. All controls to the tractor are hydraulically operated. The "V" at the front is a large, double-acting scissors that cuts horizontal lengths of cane lying close to the ground.

Front view of the unit, photo at right, shows the V-shaped cutter that follows the cane row close to the ground and oscillates fore and aft at the rate of about 100 cpm. The cane is carried over the machine on a conveyor and dumped at intervals, to form windrows perpendicular to the machine's line of travel.

14. Replanting: After the cane has been harvested and the lines reshaped, the ratoon cane is allowed to sprout for from three to five weeks. Because many of the cane stools are damaged or completely torn out, there are many blank spots in the cane rows.

This machine, known as a replanter, was developed to replace the slow job of hand planting, where the laborer carried a sack of cane seed and a pick-axe.

When the operator on either side of the tractor spots a missing cane stool, he lowers the digging shovel at the front of the tractor with his right foot to open up a short furrow. He then drops a length of cane seed into the opening. Then he covers it with the disc coverers, controlled with his left foot.

This operation is all done in second or third gear. It requires precise timing, especially where many voids are found.

This is a two-row machine, requiring a three-man crew. The cane line under the tractor remains blank until the succeeding round. The furrow openers and coverers are pneumatically raised and lowered with the help of a 20-cu ft air compressor, mounted at the front and driven from the engine crankshaft.

Paper on which this article is based is available in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.

Five High-Speed FLIGHT PROBLEMS

EXCERPTS FROM PAPER* BY

Joseph Flatt, Air Materiel Command, USAF

* Paper "Some Notes on High-Speed Flight Problems" was presented at IAS-SAE Metropolitan Section High-Speed Flight Symposium, New York, March 16, 1950.

THREE physical phenomena account for five of the worst aerodynamic troubles that plague contemporary high-speed aircraft:

1. Buffeting of the whole airplane
2. Rudder buffeting
3. Longitudinal trim changes
4. Control force variations
5. Lateral trim changes

These difficulties occur above the lowest speed where local sonic velocity is obtained at some point on the aircraft—that is, above the critical Mach number.

The three physical phenomena are (a) airflow separation at supercritical speeds, (b) loss of effectiveness of trailing-edge type of controls, and (c) dissymmetries in aircraft contours.

Airflow Separation

Fig. 1 shows that, for a typical airfoil section at Mach numbers below the critical and at positive lifts, there is practically no change in drag and a slight increase in lift and pitching moment coefficients with Mach number. At the critical Mach number and slightly above, there is little change in the variation of the coefficients, although a weak shock is present. As the Mach number is further increased, however, the shock wave increases in intensity until the boundary layer is thickened enough to cause airflow separation behind the shock. It is this separation that causes the large increase in drag and the decrease in the lift and pitching moments.

The abrupt change in drag usually occurs at Mach numbers slightly below that for the lift-moment and pitching-moment changes. At negative lifts, the moment and lift changes will be in the opposite direction. These changes are very similar to those occurring when an airfoil is stalled at low speed. In both cases, the large changes are due to airflow separation. A wide turbulent wake accompanies

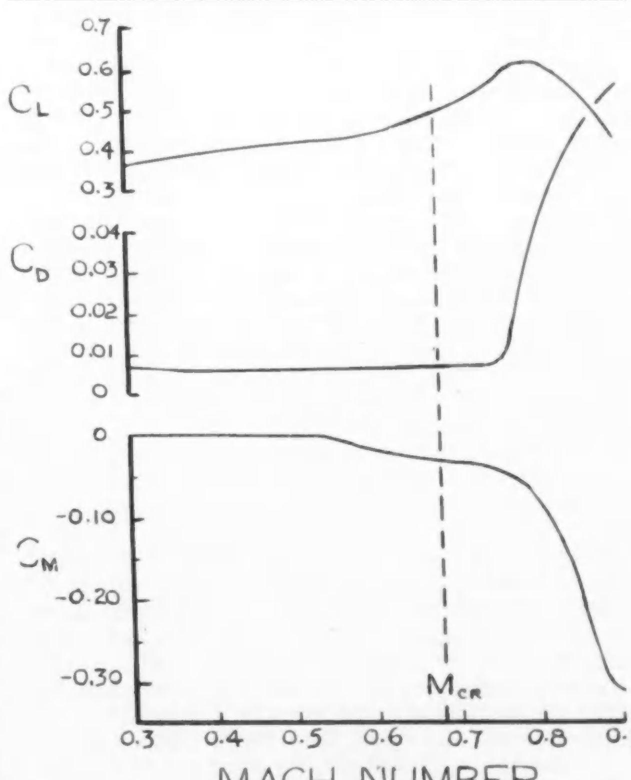


Fig. 1—Effect of Mach number on characteristics of NACA 64_1-208 airfoil when angle of attack is 2 deg

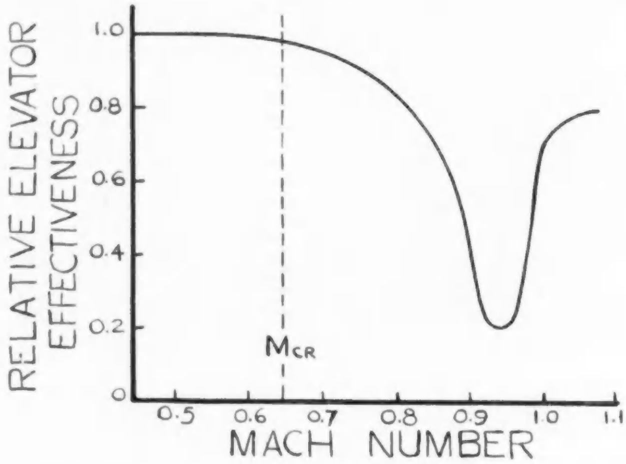


Fig. 2—Effect of Mach number on elevator effectiveness

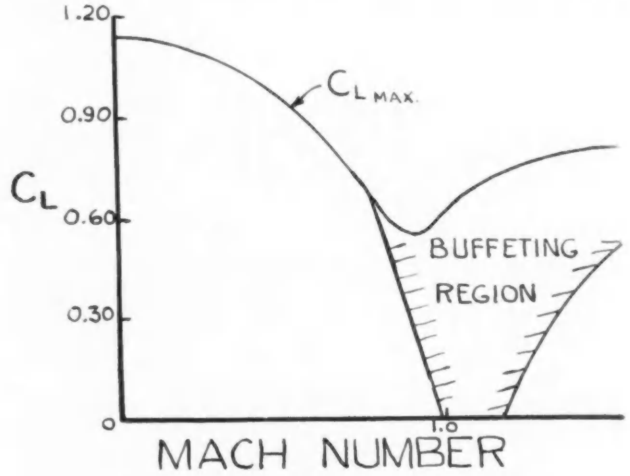


Fig. 3—Lift coefficient versus Mach number

the separation, and the downwash in the wake is considerably reduced.

Fig. 2 shows the effect of Mach number on the relative effectiveness of a flap-type control surface. At Mach numbers above the critical, it is evident that the control effectiveness is decreasing. At first the decrease is small because the pressure field created by the deflected surface can not propagate forward into the local supersonic region on the airfoil. Therefore, the effective camber change due to flap deflection is lower than at subcritical speeds. As the Mach number is further increased, this effect and the previously mentioned separation over the flap combine to cause a rapid decrease in control effectiveness. At very high Mach numbers, the shock moves downstream to the trailing edge of the wing, and the flap regains some of its effectiveness.

Of course, speed has a very large effect on aerodynamic forces. For instance, a force or moment existing at 150 mph may be small compared to the weight or moment of inertia of the aircraft, while at 600 mph this force or moment will have considerable effect on the aircraft aerodynamically and structurally. For this reason, tolerances which were acceptable on low-speed aircraft may not be satisfactory for high-speed aircraft.

1. *Buffeting of the entire airplane* is usually recognized as an unsteady variation of normal acceleration due to rapid and erratic pitching of the airplane.

The pitching is probably caused by the fluctuating wake of separated airflow impinging upon the horizontal tail surfaces. A similar condition exists at low speeds on some aircraft when the separated wake of the wing strikes the tail at or near the stall.

Exact information on the basic phenomenon involved in buffeting is practically nonexistent. It has been found, however, that the frequency of the horizontal tail in buffeting is very close to the natural frequency of the horizontal tail in bending. The indications are that most high-speed aircraft have buffeting ranges defined by Mach number and lift coefficient. A typical buffet-region diagram for a high-speed aircraft is shown in Fig. 3.

Buffeting usually occurs along the line of maximum coefficient of lift at low speeds, as Fig. 3 shows. At higher speeds, buffeting is evident at values of lift coefficient below the maximum. For a certain range of Mach numbers, it can be seen that buffeting is present at all lift coefficients. There is some evidence that the buffeting subsides at the very high Mach numbers. This is probably due to the facts that the airflow is completely supersonic and the shock wave causing boundary-layer separation no longer exists on the wing.

Rudder Buffeting

2. *Rudder buffeting*—which afflicts several of the new high-speed aircraft—is usually identified by the pilot as rudder-pedal shaking or high-frequency directional oscillation of the entire aircraft. Rudder buffeting usually occurs on those aircraft where the horizontal surface is mounted on the vertical one and is probably due to separation at the junction of stabilizer and fin.

This separation occurs at Mach numbers above the critical as determined from the characteristics of the two surfaces and their mutual interference. Because of the interference, the critical Mach number of the combination is considerably lower than that of the surfaces considered individually.

This difficulty is the result of an attempt to escape some of the other high-speed difficulties. Mounting the horizontal surface on the vertical surface alleviates tail buffeting and longitudinal trim changes, since it raises the horizontal tail higher above the downwash field of the wing. It appears, however, that in order to realize this benefit, it will be necessary to use very thin tail surfaces and carefully designed intersections.

Longitudinal Trim Changes

3. *Longitudinal trim changes* involve longitudinal stability, downwash changes, pitching moment changes, and lift changes.

Fig. 4 shows a typical example of stick force required to trim in level flight versus Mach number. As speed is increased beyond a Mach number of about 0.5, an increasing push force is required to trim. At higher Mach numbers, however, this variation becomes unstable—that is, the airplane wants to tuck under. An increasing pull force is required to prevent a divergent entry into a dive.

The loss in lift at constant angle of attack, change in downwash, and change in pitching moment all contribute to this diving tendency. As the lift decreases, the aircraft must fly at a higher angle of attack to sustain its weight. This adds to the upload on the tail and, therefore, to the diving moment. The pitching moment, of course, increases negatively and contributes directly to the diving moment.

At positive lifts, the downwash contributes a down load to the horizontal tail forces, which in terms of pitching moment is positive. When the airflow separation over the wing occurs, the downwash decreases and the net change in tail load due to downwash is positive or up, which also contributes to the negative pitching moment.

Of the three reasons for tuck-under, the effect of downwash change appears to be the most powerful. Positioning of the horizontal tail surfaces as far away as practicable from the downwash field of the wing is desirable for reducing the tuck-under tendency.

Control Force Variations

4. *Control force variations* result from the tuck-under tendency and the loss in elevator effectiveness. Both these effects increase the elevator angle required and, therefore, the elevator force required.

At low Mach numbers, the relationship between stick force and normal acceleration is usually nearly linear and independent of speed. Above the critical Mach number, there is a sharp increase in the gradient. At supercritical Mach numbers, the stick force gradient tends to become very large, making it difficult to obtain high accelerations with reasonable stick force.

Lateral Trim Changes

5. *Lateral trim changes* are due to slight dissymmetries of the aircraft. Small differences in the wing panels may make the aircraft slightly wing heavy in one direction or the other.

Usually a small amount of trim tab or aileron rigging is all that is required to remedy the situation at low speeds. At speeds above the critical, however, the aircraft may exhibit pronounced wing heaviness. Two factors contribute to this heaviness: (1) Differences in wing construction may cause airflow separation and abrupt force changes over one wing panel sooner than the other due to the effect of the local contour deviations on the critical Mach number. (2) The unbalanced rolling moment previously taken care of by trim tab or aileron rigging may reappear when the trim tab or aileron loses its effectiveness at the supercritical speeds.

Of the factors causing these five high-speed-flight troubles, separation of the airflow at supercritical speeds is the most important. Aircraft designers increase the Mach number at which separation occurs by increasing the critical Mach number and by preventing excessive boundary-layer thickness.

The accepted methods of increasing the critical Mach number are use of thinner airfoil sections, wing sweep, and low aspect ratios. In order to make use of these methods, however, we must accept structural and other aerodynamic disadvantages. Wings with large amounts of sweep and low aspect ratio require large ground angle for take-off and landing and have relatively low values of lift-drag ratio. Wings with thin airfoil sections and sweep are relatively heavy for the required strength. Wings with sweep-back introduce stalling and longitudinal stability problems at low speeds, while sweptforward wings have undesirable lateral stability characteristics. By judiciously combining the above effects, however, it is possible to improve the high-speed characteristics of an aircraft and still retain acceptable low-speed characteristics.

Preliminary experiments have shown that suction slots can effectively reduce the thickness of the boundary layer so as to delay separation. The practicability of this scheme, however, has not yet been demonstrated outside of the laboratory.

This paper is available in full in multilithographed form from SAE Special Publications Department. Price 25¢ to members, 50¢ to nonmembers.

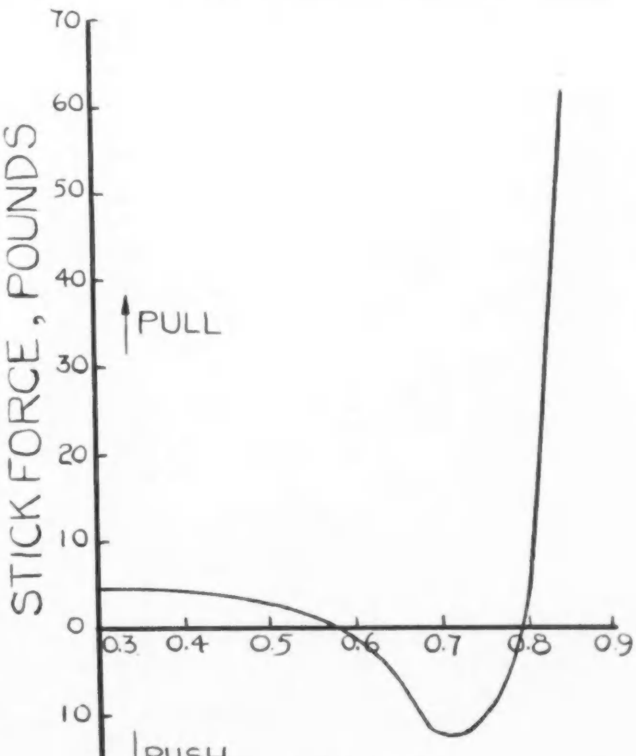
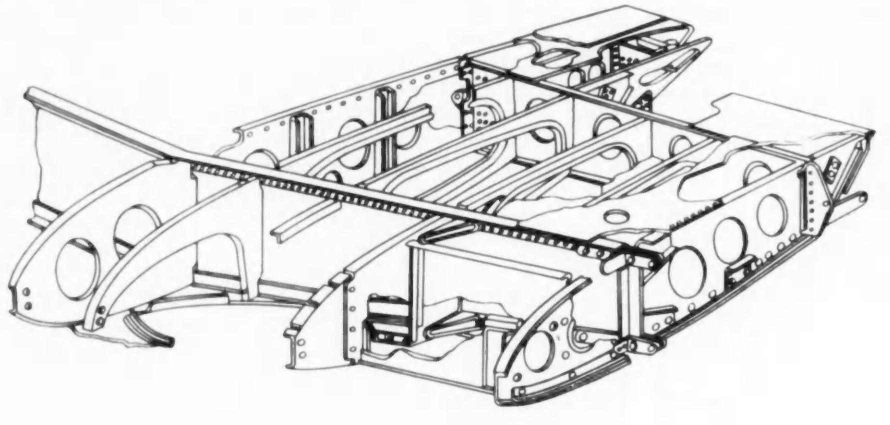


Fig. 4—Variation of stick force with Mach number



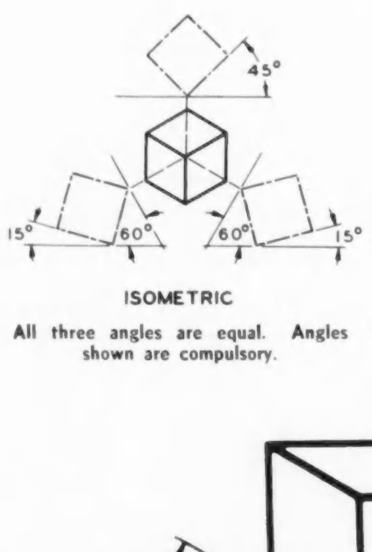
How to

PICTORIAL drawings are used in place of, or supplementary to, multi-view drawings. They have proven their worth in enabling personnel without technical training to visualize the object represented, as well as in assisting the trained designer to conceive better a new design.

In general, this type of drawing can well be applied where the problem is one of quickly conveying an overall concept of an object, rather than a strict and detailed definition or a complex geometric shape. Such applications include schematic piping and wiring systems as well as sales and field service drawings. Drawing of the aircraft wing structure above is another example. These drawings in "ex-

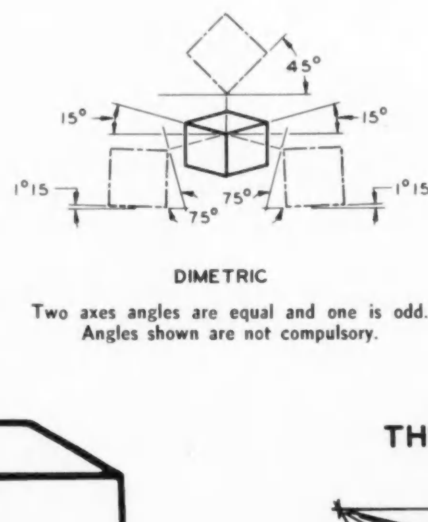
Definitions of Drawings

AXONOMETRIC DRAWINGS



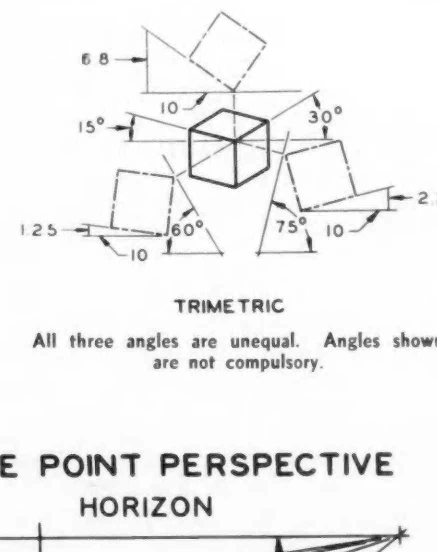
ISOMETRIC

All three angles are equal. Angles shown are compulsory.



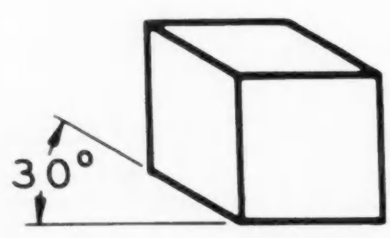
DIMETRIC

Two axes angles are equal and one is odd. Angles shown are not compulsory.



TRIMETRIC

All three angles are unequal. Angles shown are not compulsory.

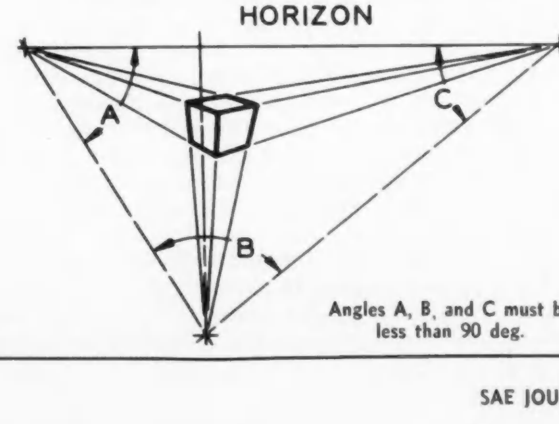


OBLIQUE

Two axes are at right angle with each other in plane parallel to the picture. The third axis may be at any angle to the horizontal line, preferably 30 or 45 deg.

THREE POINT PERSPECTIVE

HORIZON



Angles A, B, and C must be less than 90 deg.

Make and Use PICTORIAL DRAWINGS

This is a recently-approved addition to the SAE Aeronautical Drafting Manual. It was prepared by a group under the SAE Drafting Manual Committee, headed by E. O. Stark, Allison Division, GMC.

ploded" form are valuable also in describing assembly and weldment arrangements, such as powerplant and fuselage assemblies.

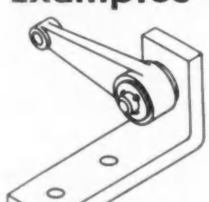
Pictorial drawings fall into these four main categories: (1) axonometric, (2) oblique, (3) perspective, and (4) free hand. Axonometric drawings include: (a) isometric, (b) dimetric, and (c) trimetric, with drafting difficulty progressing in that order.

Common forms of oblique drawing are cavalier and cabinet. Of these two, the latter usually gives a more satisfactory result. Perspective drawings are classified as: (a) one-point or parallel perspective; (b) two-point or angular; and (c) three-point. A three-point perspective drawing, while excelling all other forms of pictorial representation, is also the most difficult to construct.

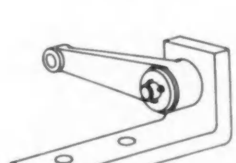
An ingenious and economical way of producing a three-point perspective drawing is the photo-bleach-out method. When pictorial drawings are "printing through process," such as blue print and ozalid, all shading—if used—should be limited to high contrast lines, block, or stipple shading. Drawings to be reproduced photographically (reflection) may be smudge, wash, or airbrush shaded.

For clarification of the major types of pictorial drawing, there follows a definition of each using a cube as the figure. Each of the various types is further illustrated by examples of a needle-bearing lever bracket assembly depicted in axonometric (isometric, dimetric, and trimetric), oblique, two and three-point perspective, exploded, and cutaway drawings.

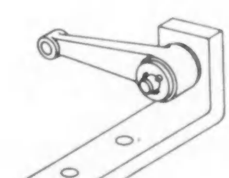
Examples of Various Methods



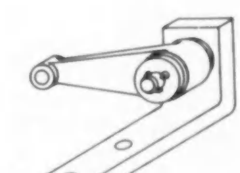
ISOMETRIC



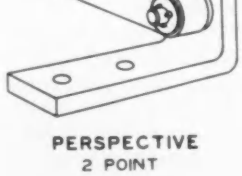
DIMETRIC



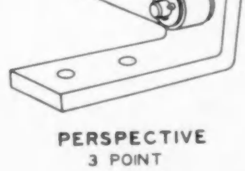
TRIMETRIC



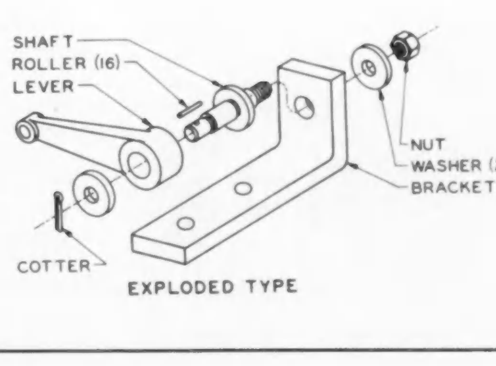
OBLIQUE



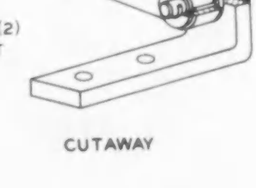
PERSPECTIVE
2 POINT



PERSPECTIVE
3 POINT



EXPLODED TYPE



CUTAWAY

Three Types of Power Steering

RUBBER-TIRED earthmovers of the self-propelled type in off-the-road operations have all had front-axle loading in recent years, making impractical manual steering. Prime movers, used with scrapers and bottom dump wagons, now incorporate power steering, of which there are three types: (1) electrical, as used in LeTourneau units; (2) mechanical with hydraulic assist, typified by Euclid machines; and (3) fully hydraulic steering systems, which Heil builds.

Power steering is a must for this heavy off-the-road equipment. As the machines get bigger, steering torque increases rapidly. Steering ratios cannot be developed within the ability of drivers to handle them properly. Reversibility of the steering gear, caused by road shock when these units traveled over rough terrain, resulted in injuries to drivers. This made imperative the improvement of steering mechanisms for these prime movers.

Fig. 1 shows a typical LeTourneau unit called the "Tournapull," which uses an electric steering method. This electrical steering is accomplished by a high torque electrical motor, which operates from the current supplied by an engine-driven generator. These electric motors, referred to as "Tournatorque Motors," are completely designed and built by the R. G. LeTourneau Co.

They are a-c motors built with the lugging characteristics of a d-c, specially designed for the intermittent load service required for construction work. They give full, smooth, instantaneous power for the short periods required for turning. Special thermal-type switches prevent overheating; and built-in limit switches automatically cut off power the instant extreme steering position is reached.

These Tournatorque motors brake instantly, lock and hold by means of special disc-type brakes, and are immediately reversible. These motors have but one wearing part—the brake lining. One of the most important improvements in this Tournapull with this method of steering is that it eliminates the danger of jack-knifing when traveling down steep grades.

Fig. 1—The LeTourneau Tournapull is steered electrically

Fig. 2 is a cross-section view showing the complete generator drive and location of the steering mechanism of the Tournapull. Some of the principal parts of the steering mechanism are the king pin housing, the steering gear housing, steering motor with brake, a gear reduction box, ring gear, and king pin. The motor, gear box, and pinion are fixed with relation to the scraper.

The steering motor, which is bolted to the scraper, meshes with a ring gear which is fastened to the prime mover. The king pin and ring gear are fixed in relation to the Tournapull. Rotation of the steering motor causes the prime mover to turn to the left or to the right with respect to the scraper.

When a LeTourneau operator wishes to make a left turn, he moves the steering switch located on the panel to the left. This causes a contactor switch to close, permitting power to flow to the winding of the steering motor, causing it to rotate. The electro-magnets which release the steering motor brake are also energized by this current.

Since rotational speed of the steering motor is too fast for the steering needs, the steering gear reduction box reduces the speed of the drive pinion; at the same time it greatly increases the torque. The electric hook-up is such that the gear reduction box pinion turns the steering ring gear counter-clockwise with relation to the king pin centerline when viewed from above. Obviously counter-clockwise movement of the ring gear changes the directional course of the prime mover to the left.

To go to the right, the steering switch is moved to the right, causing the electrical circuits to reverse the rotation of the steering motor. This in turn causes the ring gear to be moved clockwise (looking down on it), changing the direction of the prime mover to the right.

The steering switches are spring loaded. When released, they automatically return to the "off" position, locking the prime mover at whatever angle it happens to make with the scraper at that instant. This "locking" effect comes about through the steering motor brake, which immediately prohibits movement of the rotor as soon as the current is turned off. The precision fit of the entire chain of gears

For Off-the-Road Earthmovers

EXCERPTS FROM PAPER* BY

G. J. Storatz, Engineer-in-Charge, Road Machinery Division, The Heil Co.

* Paper "The Steering of Rubber-Tired, High-Speed, Off-the-Road Earthmoving Units," was presented at the Earthmoving Industry Conference, SAE Central Illinois Section, Peoria, April 12, 1950.

from the steering motor to the steering ring gear prevents any free play or slack in the prime mover with the steering motor brake on.

The degree of turning in either direction is controlled by a limiting switch which controls the steering motor and which can be set for any angle up to slightly more than 90 deg.

As an added safety feature, a prominent warning light has been placed on the dash panel. It will indicate trouble in the steering mechanism sufficiently in advance to allow the operator time to stop.

The second steering method is the mechanical type with hydraulic assist. It is used on the Euclid unit, a bottom dump wagon with a four-wheel type of prime mover in which the two front wheels are used for steering. See Fig. 3.

On the larger Euclid machines, a Ross-Bendix hydraulic steering gear assembly is used. The steering linkage is of conventional design.

Fig. 4 shows the installation of this gear on the euclid tractor. This hydraulic power steering gear incorporates the Ross cam and lever gear design with the power cylinder and control valve supplied by Bendix. It is a compact, completely-housed, integral assembly, and is installed with the same type of mounting used with conventional steering gears.

The mechanical linkage between the steering wheel and knuckles is similar to that used in a mechanical steering mechanism. This integral type of hydraulic power steering gear provides more than finger tip control. It provides an effortless and fatigueless steering with the added factor of greater safety because it provides that same quality of road sense steering obtained with the conventional manually-operated steering gear.

Ability of the steering wheels to return to straight ahead driving position, after making a turn, is an extremely valuable characteristic of this design. Loss of control in soft ground, sand and snow, or from tire blowouts or any other road obstructions, is instantly prevented by automatic operation of the unit. There is no lag in the hydraulic response either for power assistance or resisting shocks. Thus, there is no tendency to over control.

The mechanism that transmits hydraulic power to the steering gear consists of a hydraulic pump and reservoir, control valve, and power cylinders. Hydraulic power is applied to the cross shaft through an extension of the lever, which contacts a sliding block connected to the piston rod of the power cylinder. Flow of oil to the cylinder is directed by the control valve. The oil itself is supplied by an external pump which is driven by the engine. A relief valve is provided and is set for 750-psi maximum pressure.

Shown in phantom in Fig. 5, are the principal working parts of the gear. They are the cam, lever-shaft, sliding block, control valve, and power cylinder.

Action of this steering gear is both manual and hydraulic in effect. When the cam is turned to the left or right, by the driver's effort on the steering wheel, this stud of the lever is moved through the groove of the cam. This turns the lever shaft and provides angular movement of the steering arm. However, whenever the driver's effort at the steering wheel exceeds the pre-load of the control valve centering springs, the hydraulic system comes into operation automatically and relieves the driver of excessive loads.

In addition to acting as a booster, the hydraulic system resists kickbacks or shock, which might otherwise cause the driver to lose control of the vehicle.

Fig. 6 shows the valve which is the heart of the hydraulic system. This is a sectional drawing of the valve assembly and shows the three valve positions and fluid flow through the valve. The valve housing has several internal oil channels, through which the oil flow is permitted or restricted, according to the position of the valve spool.

The spool moves axially with the cam. These two parts are restrained from axial movement by the pressure of the centering springs and the oil pressure against the centering plungers. (See Fig. 6A.) Both effects tend to center the spool and thus hold the valve in the neutral position. The inner portion of the centering plungers contact the thrust washers at both ends of the valve assembly. The outer por-

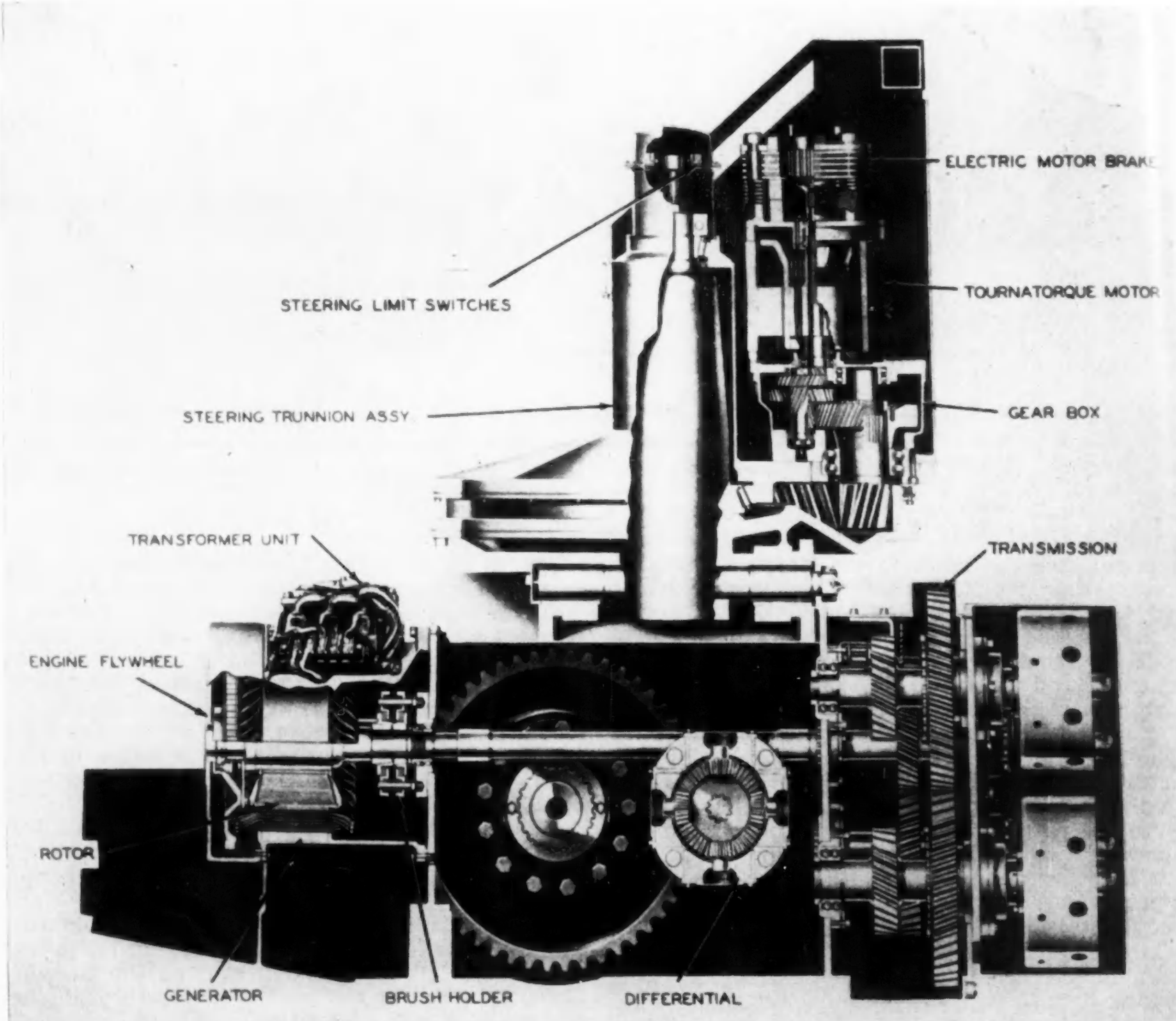


Fig. 2—Cross-section of generator and steering motor for the Tournapull earthmover

tion of the plungers contact the valve adapter at one end of the valve and the end cap at the other end.

Axial movement of the cam compresses the centering springs between one thrust washer and adapter in one direction and the other thrust washer and end cap in the opposite direction.

Whenever the driver's effort at the steering wheel overcomes the centering effect of the valve springs, the valve spool is moved axially. This restricts both return passages to the outlet port, however, leaving the outlet port open to one end of the power cylinder. It increases pressure at one of the cylinder ports. The immediate effect is high hydraulic pressure in an end of the cylinder to actuate the piston, which applies hydraulic power directly to the inner lever of the gear.

Figs. 6B and 6C illustrate full valve movement in both directions. Full pressure is obtained with only a few thousandths of an inch travel. The slightest movement will result in a pressure differential. Whenever the gear is subjected to shock loads, the hydraulic system functions similarly, thus preventing a kickback at the steering wheel. Whenever the effort at the steering wheel is released, the valve spool is returned to the neutral position.

When the valve is in the neutral position, with the spool centered, the oil pressure at its two cylinder ports is low. This pressure is equal and produces ineffective pressures on the piston in the cylinder. There is no movement of the piston from pump pressure and no circulation of oil in the lines to the cylinder. After a turn however, the self-righting effect of the front wheels will return the steering gear and piston to the straight ahead position. While this is happening, there is a slight movement of oil in the cylinder lines which are under the equalized low pressure.

When the valve is in the neutral position steering action is strictly manual in effect. This gives the driver the feeling of "road sense." This feeling, which permits the sense of touch to assist the sense

Fig. 3—The Euclid four-wheel prime mover, shown here with a bottom dump wagon, is steered by a mechanical system with hydraulic assist

of sight, is fully appreciated by experienced drivers and is desirable for several safety reasons.

The third type of power steering is the full hydraulic. It is the most common type of steering that is being used exclusively on the two-wheel type of prime movers. Such a system is used in steering the two-wheel type of Heiliner, shown in Fig. 7.

The Heil Hydro-steering system on these vehicles consists of a constant delivery engine-driven hydraulic pump, a steering head to which the steering wheel is attached, an oil reservoir equipped with a full flow filter, and two hydraulic double-acting cylinders.

Fig. 8 shows the hydraulic constant pump which is located on the front end of the crankshaft. It delivers a constant volume of oil to the Hydro-steer at all times, regardless of engine speed. The pump is of a vane-type construction. It has 15 vanes of hardened and ground alloy steel, hydraulically balanced for smooth, nonpulsating flow of oil. This pump has a floating control valve ring which regulates automatically, delivering a constant flow of oil to the system.

The delivery of the pump is set by a metered orifice in the outlet port. The constant delivery of this pump is controlled by a spring-loaded piston, which

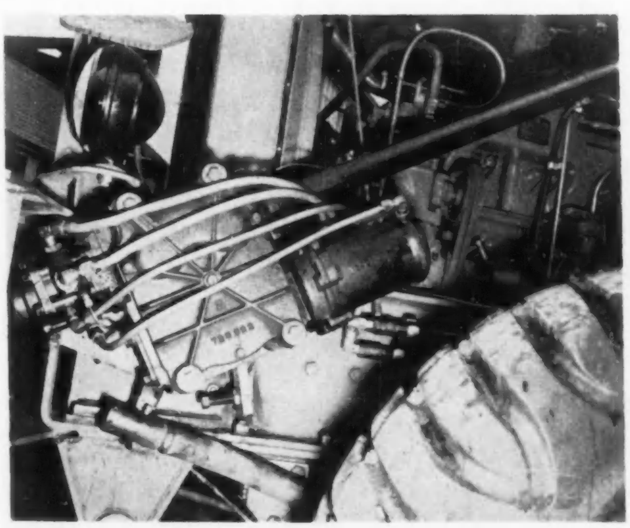


Fig. 4—Installation of the hydraulic steering gear on the Euclid prime mover

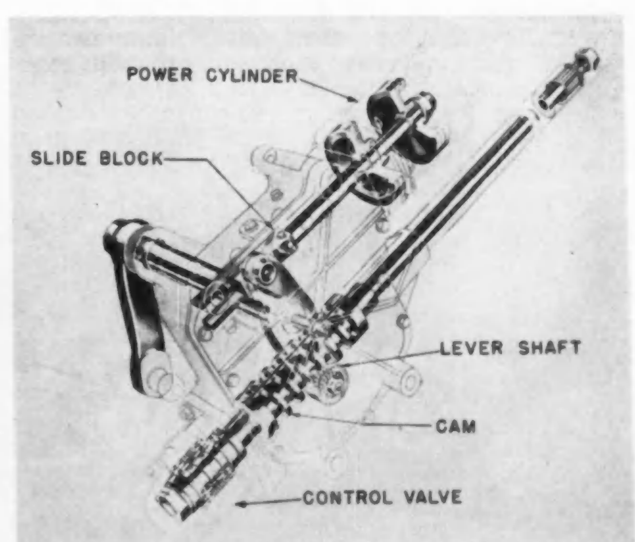


Fig. 5—Phantom view of the mechanical-hydraulic steering system for steering the Euclid unit

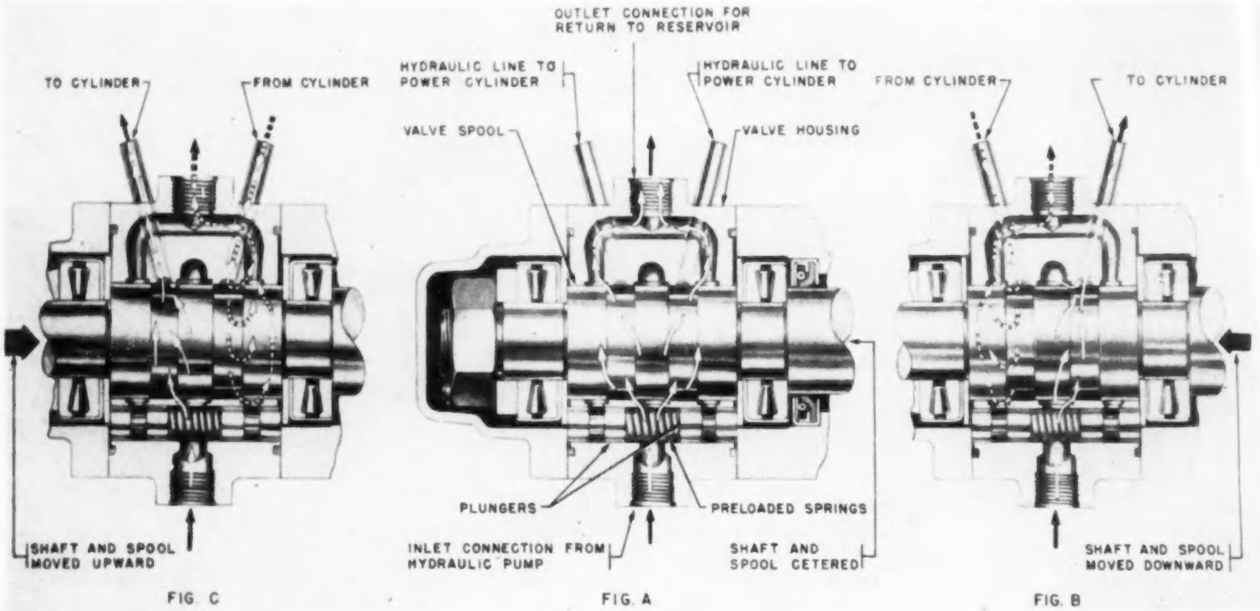


Fig. 6—Control valve positions for the three operating conditions of the Euclid machine's steering system. "A" shows the neutral valve position. "B" shows valve position for a right turn. "C" shows valve position for a left turn. White arrows represent fluid flow from pump; dotted arrows, displaced fluid from cylinder

moves the regulating ring to correspond to the back pressure created by the variations of the rpm.

The faster rpm tends to deliver more oil; but by being restricted through the metering orifice, this volume of oil creates a back pressure. When acting on this plunger, the back pressure causes it to move and position the regulating ring so that the delivery drops down to the required gallons per minute.

When the rpm goes down, the opposite happens and the ring moves in the opposite position, allowing increased flow of oil. At all times, whether the engine is turning over at 600 rpm or 2100 rpm, approximately the same amount of oil is delivered to the system.

Fig. 9 shows the steering head. It is similar in principle and construction to one for an earlier system for a Heil four-wheel unit. Requirements of the two-wheel unit make it necessary that the steering mechanism have about five times the capacity of the one for the four-wheel prime mover.

Major parts of the steering head are: the spool valve, located in the front part of the head; the servo mechanism operating the spool valve; a check valve; and a set of metering gears located in the bottom of the steering head. Even though there is a reservoir in the head of the steer, an auxiliary tank is used to allow for a larger volume of oil to keep down oil temperature. A pressure relief valve is attached directly to the side of the oil tank in the pressure line coming from the pump to the head.

The spool valve in the head is spring loaded. This keeps it in neutral position when the vehicle is moving forward. Oil from the engine-driven pump is circulated from the reservoir part of the steering head back into the center port of the head, through the drilled holes in the spool valve, and back into

Fig. 7—The Heil two-wheel prime mover has a completely hydraulic steering system

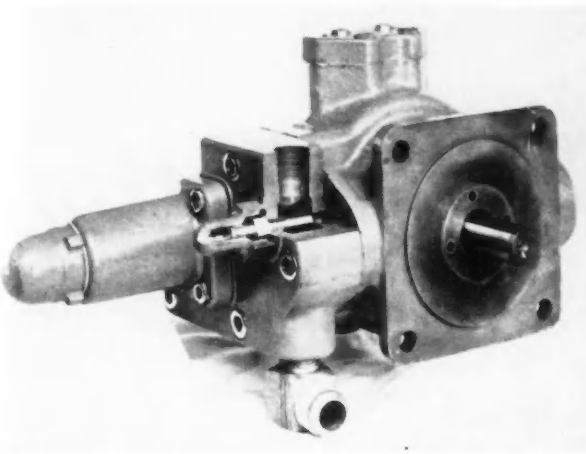


Fig. 8—This pump for the Heil full hydraulic steering system delivers a constant volume of oil, regardless of engine speed

the reservoir part of the head. This oil is circulated at very low pressure, and consequently no power is lost.

Steering of the Heiliners is accomplished by the action of two interconnected double-acting cylinders, acting on the fifth wheel spindle housing which is part of the trailing vehicle. The cylinders themselves on the rear end are attached to the fifth wheel spindle mast assembly, which is a part of the tractor. The steering cylinders push the two-wheel tractor to the right or left in relationship to the trailing member of the unit.

Inherent in the design of the unit is the positioning of the spindle in such a manner that when the vehicle turns to the right or to the left, it tends to have an action very similar to the camber and caster as designed in passenger cars. This causes a positive control of the steering without any tendency of hunting, even if there is any slack in any of the linkage points.

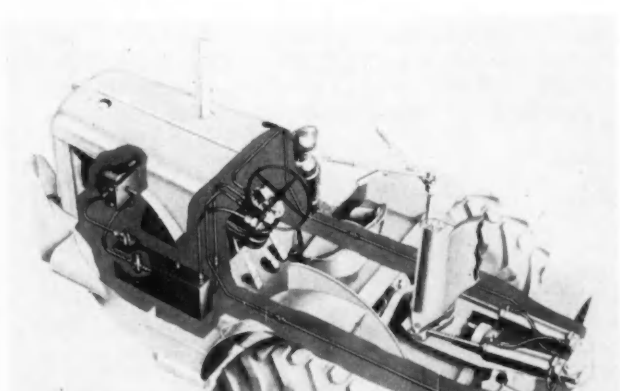


Fig. 10—Piping diagram of the completely hydraulic steering system for the Heil two-wheel prime mover

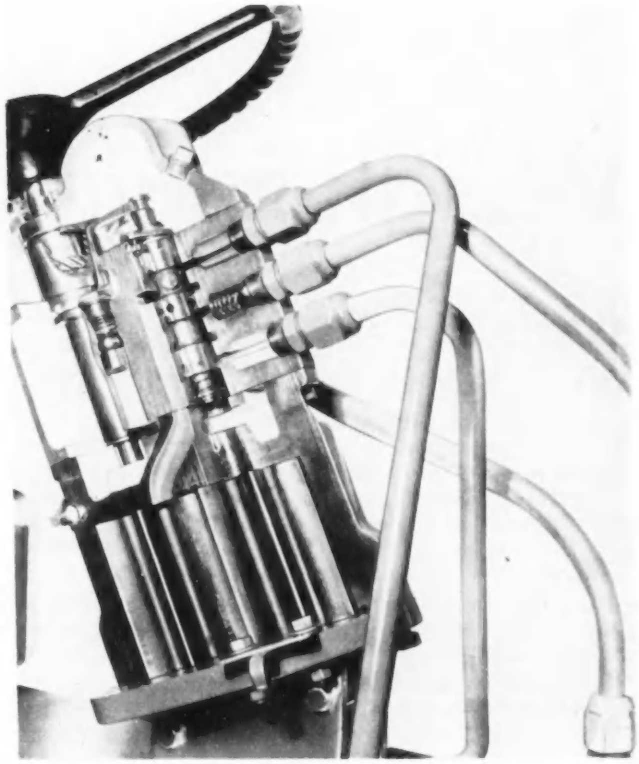


Fig. 9—Steering head for the hydraulic steering system of the Heil two-wheel prime mover is the heart of the system

Fig. 10 is a schematic view of this hydraulic system. When the steering wheel is in the neutral position, the oil flow is as follows:

Oil from the reservoir in the steering head returns to the oil reservoir, which is attached to the frame. This oil then passes through the filter in the reservoir taking out all impurities—even those microscopic in proportion. The clean oil then flowing through the suction line to the suction side of the pump. From the pump, the oil is forced through the pressure line through a relief valve, and then through the center port of the valve and back into the oil reservoir. This circuit is continuous and operates at hardly any pressure when the vehicle is not being steered.

When the steering wheel is turned to the left, the steering valve and the servo mechanism force the oil to feed through the metering gears and out of the top port, into the hydraulic line.

The servo mechanism, operating a helical spiral cam mechanism, moves the plunger valve by positioning it in such a way that the oil, coming from the pump into the center port, is directed down into the metering gears.

These metering gears are motorized and rotate at a pre-determined speed, regulating the rate of steer. The oil then is discharged from the opposite end of the metering gears, back through the head and through the top port, into the hydraulic line.

The oil then is forced under pressure through the hydraulic line feeding into the bottom of the left-hand cylinder and the top of the right-hand cylin-

Continued on Page 61

Weigh Engine Wear Factors

EXCERPTS FROM PAPER* BY

W. S. James and B. Gratz Brown

Vice-President of Engineering, Fram Corp.

Chief Engineer, Dexter Division, Fram Corp.

* Paper "Engine Wear as Affected by Air and Oil Filters," was presented at SAE Summer Meeting, French Lick, Ind., June 6, 1950.

TWELVE tests run with a car in the Fram dust tunnel, at Dexter, Mich., to evaluate various sources of engine-damaging dirt led to these 10 conclusions:

1. Quantitative evaluation can be made of factors causing engine wear and means to prevent it. This will help the design engineer determine how much of his protection dollar he should spend on which wear cause, and effectiveness of available protection devices.
2. Some oil-wetted type carburetor air cleaners provide almost no engine protection against airborne dust.
3. An oil-bath carburetor air cleaner will reduce ring and bore wear to one-tenth that with some oil-wetted types.
4. Using an oil filter, when properly serviced, will reduce bearing wear by 50% or more and definitely reduces oil ring wear.
5. The "pull-over" of oil-bath carburetor air cleaners is very serious and may reduce life by several times.
6. Increasing efficiency of a carburetor air cleaner from 98 to 99% halves wear.
7. Chrome-plated top rings will double the time the engine can be run until excessive blowby occurs.
8. Chrome-plated top rings will reduce bore wear by about 75%.
9. Chrome-plated top rings may not reduce wear of the lower plain rings.
10. Increasing chrome plate thickness on the top ring increases ring life, but not in the same proportion.

The protective measures used in each of the tests are shown in Table 1.

The first test was made with no protection on the engine. It was not as complete as later tests since we did not know how the equipment would work, how long the engine would run, how to tell when the test should be stopped, and so forth. The next four tests compared the relative life of the engine with oil-bath and oil-wetted types of carburetor air cleaners, when used with and without oil filters. The sixth test was to check the performance of what we believed to be a high efficiency replaceable cartridge type of carburetor air cleaner.

The next four tests were made with the cooperation of a piston-ring manufacturer on the effect of chrome-plated top rings on engine and ring life. The eleventh test was to check an improved type of high efficiency replaceable cartridge type of air

Table 1—Dust Tunnel Test Program

TEST	No. 1	No. 2	No. 3	No. 4	No. 5	No. 6	No. 7	No. 8	No. 9	No. 10 (Repeat No. 1)	No. 11	No. 12
Carburetor Aircleaner	No	Oil Wetted	Oil Wetted	Oil Bath	Oil Bath	Replaceable Type A	No	No	No	No	Replaceable Type B	Oil Bath
Oil Filter	No	No	Yes	Yes	No	Yes	No	No	No	No	Yes	No
Crankcase Aircleaner	No	Metal Mesh	Metal Mesh	Metal Mesh	Metal Mesh	Metal Mesh	No	No	No	No	Metal Mesh	100%
Gas Filter	No	No	No	Yes	Yes	Yes	No	No	No	No	Yes	Yes
Top Compression Ring	Production	Production	Production	Production	Production	Production	2/4 Chrome 3 cyl	4/6 Chrome 3 cyl	4/6 Chrome 6 cyl	Production	Production	Production Special Ventilation

In MAN-MADE Dust Storm

cleaner. The twelfth test was to show the importance of clean air in the crankcase ventilating system.

Results of the first series of tests with an oil-wetted type of carburetor air cleaner are given in Table 2. First column of the table gives the results of a run with no protection on the engine. In the second column are the results with an oil-wetted type of carburetor air cleaner, and a wire mesh crankcase air cleaner, but with no oil filter. In the third column are the results with an oil filter added.

The results of a run with no protection are from test No. 10. These are believed to be more reliable than those obtained in test No. 1 because more experience had been gained by that time.

Comparison of the wear under these three conditions indicates that the piston-ring wear was very rapid, being over ¼ in. on the top ring gap in all cases. The order of decreasing wear of the several rings was the same in all tests—top compression, top oil, second compression, and then second oil. This order of decreasing wear is different from that usually encountered with passenger cars in normal service, which usually is: second oil ring, first oil ring, top compression ring, and second compression ring. However, tests with military vehicles during the war under very dusty conditions showed the same ring wear pattern as was found in the dust tunnel.

This pattern is consistent with the fact that large amounts of dust enter the engine through the carburetor. The second compression ring is protected

Table 2—Effectiveness of Oil-Wetted Air Cleaner and Oil Filter

Test Number	No. 10	No. 2	No. 3
Carburetor Air Cleaner	None	Oil Wetted	Oil Wetted
Oil Filter	None	None	Yes
Crankcase Air Cleaner	None	Wire Mesh	Wire Mesh
Total Test Time—Hours	4.5	5.3	5.2
Average Dust, per cu ft	0.011	0.0147	0.0115
Blowby—End of Test	8.2	8.2	8.1
Average Compression Loss, psi	28	48	42
Piston-Ring Gap Increase			
Top	0.296	0.282	0.251
Second	0.172	0.149	0.133
1st Oil	0.206	0.179	0.151
2nd Oil	0.163	0.141	0.125
Cylinder Bore Increase—Thrust Axis			
Down from top ½ in.	0.0089	0.0076	0.0084
¾ in.	———	0.0060	0.0070
1 in.	0.0070	0.0049	0.0057
2 in.	0.0045	0.0032	0.0039
3 in.	0.0041	0.0029	0.0032
4 in.	0.0035	0.0029	0.0034
Bearing Clearance Increase			
Rods	0.0005	0.0005	0.000
Mains	0.0008	0.002	0.001
Bearing Wt. Loss—Grams			
Rods	0.336	0.390	0.207
Mains	0.550	0.661	0.474
Journals—Diametral Decrease			
Rods	0.0000	0.0000	0.0000
Mains	0.0005	0.0005	0.0005
Crankcase Oil Analysis—End of Test			
Pet. Ether Insolubles—% Volume	4.50	5.00	3.00
Chloroform Insolubles—% Volume	3.00	3.00	2.00
Chloroform Solubles—% Volume	1.50	2.00	1.00

Table 3—Effect of Oil Filter and Oil-Bath Air Cleaner "Pull-Over"

Test Number	No. 5	No. 4	No. 6
Carburetor Air Cleaner	Oil Bath	Oil Bath	Disposable
Oil Filter	No	Yes	Yes
Crankcase Air Cleaner	Metal Mesh	Metal Mesh	Metal Mesh
Total Test Time, hr	25	25	25
Average Dust, g per cu ft	0.0144	0.0123	0.0130
Blowby—End of Test, cfm	5.0	3.0	2.0
Average Compression Loss, psi	26	13	13
Average oil pressure loss, psi	63	——	9
Piston Ring Gap Increase			
Top	0.140	0.128	0.021
Second	0.063	0.048	0.009
1st Oil	0.121	0.074	0.019
2nd Oil	0.098	0.056	0.014
Cylinder Bore Increase—Thrust Axis			
Down from top ½ in.	0.0044	0.0043	0.0011
¾ in.	0.0033	0.0015	0.0008
1 in.	0.0022	0.0010	0.0006
2 in.	0.0013	0.0000	0.0007
3 in.	0.0013	−0.0001	0.0007
4 in.	0.0009	0.0000	0.0006
Bearing Clearance Increase			
Rods	0.001	−0.0003	0.0000
Mains	0.0015	−0.0002	0.0006
Bearing Weight Loss, grams			
Rods	0.729	0.172	0.118
Mains	0.896	0.601	0.154
Crankshaft Journals—Diametral Increase			
Rods	0.0000	0.0000	0.0000
Mains	0.0005	0.0000	0.0000
Crankcase Oil Analysis—End of Test			
Petroleum Ether Insolubles, % Volume	5.00	0.20	Trace
Chloroform Insolubles, % Volume	2.50	0.02	——
Chloroform Solubles, % Volume	2.50	0.18	——

by the top ring and the second oil ring is protected by the top oil ring. Possibly some of the dirt-carrying oil is drained off by the slots in the top oil ring, and the second oil ring is supplied by somewhat cleaner oil from the crankcase. In the case of the engine running in lower dust concentrations, the higher unit pressure on the surfaces of the oil rings may more than offset the effect of dirt reaching the compression rings.

The cylinder bore wear shows some improvement from the use of the wetted type of air cleaner, but is not consistent with the ring wear. The bearing and journal wear shows an improvement due to the use of an oil filter. But the high percentage of chloroform insolubles in the used oil at the end of the test where the oil filter was used, indicates that it was loaded to capacity. It should have been changed at some time during the test. This conclusion is borne out by the results of later tests.

In Table 3 the wear results of three tests are presented which compare the effect on engine wear of the use of an oil filter when good carburetor air cleaner protection is provided. It also shows the effect of "pull-over" in an oil bath air cleaner. A comparison of the wear results given in Table 3 with those of Table 2 indicates the great improvement in wear reduction with the oil-bath air cleaner over the oil-wetted type. In the case of the oil-bath type of carburetor air cleaner, the test ran five times as long—25 instead of 5 hr—and the wear was about half as much, an overall difference of about ten to one. The blowby at the end of the test was much lower.

The wear pattern of the rings is also changed, being more like that of normal clean road use. The second compression ring now has the least wear, instead of the bottom oil ring, so that the effect of high unit pressure on the oil ring is beginning to more than offset the greater wear from dust.

Test No. 5 was stopped because of a bearing knock. A comparison with test No. 4 shows very clearly the increase in wear from the omission of an oil filter. The effect of leaving off the oil filter in test No. 5 not only increased the wear on the rod bearings between three and four times, but it almost doubled the wear on the oil rings and the lower portion of the cylinder bore. The condition of the oil was much poorer, as shown by the chloroform insolubles, which increased more than one hundred times.

Probably the oil filter cartridge should have been changed before the end of test No. 4 because it had collected almost as much weight of dirt as was collected in test No. 3 with the oil-wetted type of carburetor air cleaner. But in test No. 6, where the

d entering the engine was less, only about half a uch dirt was collected in the oil filter cartridge; a the condition of the oil at the end of test No. 6 w the replaceable filter element was almost perf , at least as indicated by the chloroform insolub when only a "trace" was found.

he figures in the last two columns of Table 3 s w the effect of "pull-over" of an oil-bath carb retor air cleaner on engine wear. The results of te ts Nos. 4 and 6 show only the effect of a change in the carburetor air cleaner. The one used in test No. 6 was of the disposable cartridge type. Although it had the same efficiency as the oil bath type on the carburetor test bench, it showed a marked reduction in engine wear of the piston rings and cylinder bore.

In the case of the piston rings, the wear is reduced to from one-fourth to one-fifth, and to about one-half in the case of cylinder bore. There was a definite reduction in loss of weight of the bearings with the disposable carburetor air filter. This was probably due to the need for an oil filter cartridge change in test No. 4 as indicated by the greater amount of material insoluble in chloroform.

Because of the important effect which chrome-plated top compression rings were known to have on ring and bore wear, it was decided to see what results could be obtained with such rings in the dust tunnel. With this in mind, tests Nos. 7, 8, 9, and 10 were planned and carried out in cooperation with Mr. Paul S. Lane of the Muskegon Piston Ring Co.

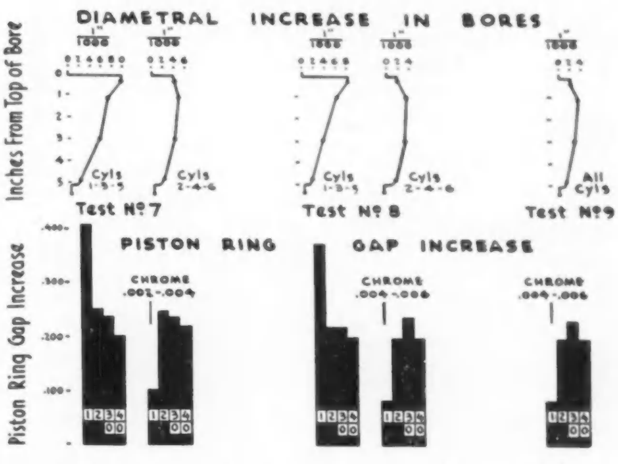


Fig. 1—Effect of chrome-plated top ring on ring and bore wear. Chrome-plated top compression rings in cylinders 2, 4, and 6, in tests Nos. 7 and 8, produced less ring and bore wear than the plain iron rings in cylinders 1, 3, and 5

In tests Nos. 7 and 8, the top compression rings

How the Dust Tunnel Works

In the Fram dust tunnel it is possible to operate a motor vehicle in a controlled dust storm. The dust tunnel is a room 50 ft long, 15 ft wide, and 13 ft high. A 200,000-cfm capacity blower is mounted slightly below the floor level in one end of the room. It is driven by V-belts from one of the two long 16-in. diameter drums located transversely across the room. These drums are driven by the rear wheels of the car under test. Cross-section of the arrangement is shown below at left. Plan view of tunnel and control room is shown at right.

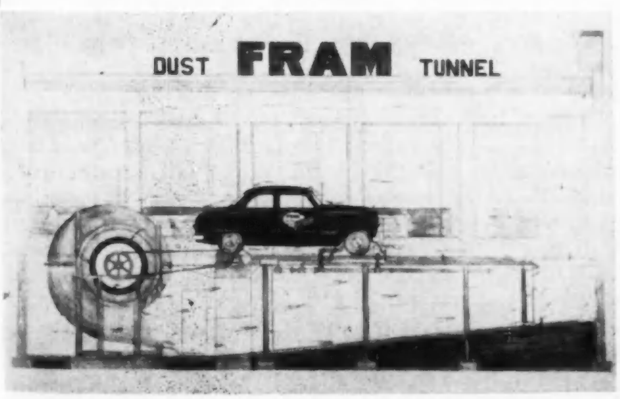
The blast from the blower passes forward through a duct under the working floor. It is guided back toward the car and into the blower inlets by a set of circular guide vanes mounted at the other end of the room from the blower. Dust is fed into the blower inlet and is blown around with the air blast and directed at the front of the car.

Controls are located in a room adjacent to the dust tunnel and separated from it by large glass panels. Operation of the car in the tunnel—except for starting the engine—is remotely controlled from the control room.

The car and blower are started and brought up to full speed before dust is introduced. Enough dust is fed to give a dust concentration of 0.0125 g per cu ft of air.

Running is continued until a shutdown is necessary due to an air cleaner or carburetor becoming plugged, or some other condition makes it impossible to continue running at full load and desired speed—60 mph on the tests to date. A continuous period of operation has rarely exceeded 2 hr, which nearly empties the gasoline tank.

Length of test depends largely on the degree of filter protection afforded the engine. End of test may be determined by high blowby, low compression pressure, low oil pressure, or mechanical failure.



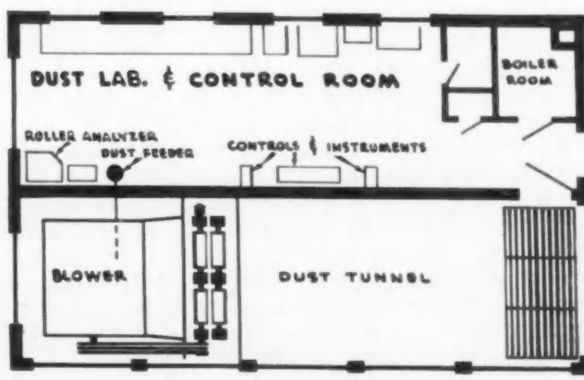


Table 4—Comparison of the Performance of Two Air Cleaners

Test Number	No. 6	No. 11
Carburetor Air Cleaner	Disposable Type A	Disposable Type B
Oil Filter	Yes	Yes
Crankcase Air Cleaner	Metal Mesh	Metal Mesh
Total Test Time, hr	25	25
Average Dust, g per cu ft	0.0133	0.0125
Blowby—End of Test, cfm	2.0	1.5
Average Compression Loss, psi	14	13
Average Oil Pressure Loss, psi	9	0
Piston Ring Gap Increase		
Top	0.021	0.011
Second	0.009	0.004
1st Oil	0.019	0.009
2nd Oil	0.014	0.007
Cylinder Bore Increase—		
Thrust Axis		
Down from top ½ in.	0.0011	0.0005
¾ in.	0.0008	———
1 in.	0.0006	0.0003
2 in.	0.0007	0.0003
3 in.	0.0007	0.0002
4 in.	0.0006	0.0002
Bearing Clearance Increase		
Rods	0.0000	0.0001
Mains	0.0006	0.0001
Bearing Weight Loss, grams		
Rods	0.118	0.046
Mains	0.154	0.145
Journals—Diametral Increase		
Rods	0.0002	0.0000
Mains	0.0001	0.0002
Crankcase Oil Analysis—		
End of Test		
Petroleum Ether Insolubles, % Volume	Trace	0.05
Chloroform Insolubles, % Volume	"	Trace
Chloroform Solubles, % Volume	"	0.05

Table 5—Comparison of Performance of Oil-Wetted Metal-Mesh Crankcase Air Cleaner and Perfect Crankcase Air Cleaner

Test Number	No. 5	No. 12
Carburetor Air Cleaner	Oil bath	Oil Bath
Oil Filter	No	No
Crankcase Air Cleaner	Metal Mesh	100% air filter
Total Test Time, hr	25	25
Average Dust, g per cu ft	0.0144	0.0123
Blowby—End of Test, cfm	5.0	1.5
Average Compression Loss, psi	26	13
Average Oil Pressure Loss, psi	63	0
Piston Ring Gap Increase		
Top	0.141	0.141
Second	0.063	0.070
1st Oil	0.121	0.125
2nd Oil	0.098	0.093
Cylinder Bore Increase—		
Thrust Axis		
Down from top ½ in.	0.0044	0.0052
¾ in.	0.0033	———
1 in.	0.0022	0.0025
2 in.	0.0013	0.0018
3 in.	0.0013	0.0016
4 in.	0.0009	0.0015
Bearing Weight Loss, grams		
Rods	0.729	0.265
Mains	0.997	0.394
Bearing Clearance Increase		
Rods	0.0013	0.0003
Mains	0.0015	0.0005
Crankcase Oil Analysis—		
End of Test		
Petroleum Ether Insolubles, % Volume	5.00	5.5
Chloroform Insolubles, % Volume	2.50	3.2
Chloroform Solubles, % Volume	2.50	2.3

on the pistons in cylinders Nos. 1, 3, and 5 were left unchanged, and those in cylinders 2, 4, and 6 were chrome plated. In test No. 7, the thickness of the chrome plate was from 0.002 to 0.004 in., and in test No. 8 from 0.004 to 0.006 in. thick. In test No. 9, all top rings were chrome plated to a thickness of 0.004 to 0.006 in. In test No. 10 no chrome was used and the rings were of production specification.

In all the tests of this series, no protection was used on the engines. In all cases, at the end of the test only a very small section of the top compression rings which had been chrome plated still had traces of chrome.

The essential wear figures are shown graphically in Fig. 1. The wear on cylinder walls is shown in the upper part of the figure and the increase in piston-ring gap is shown in the lower portion. The diametral increase in the bores is shown by the horizontal distances; the position in the bore from the top of the block is shown in the vertical scale.

The bore wear pattern in the cylinders with unchromed top rings—1-3-5—is very similar in both tests Nos. 7 and 8. Also the bore wear pattern in cylinders 2, 4, and 6 in test No. 8 and all cylinders in test No. 9 are almost identical. These bores had pistons with chrome-plated rings with from 0.004 to 0.006 chrome. Compare the wear pattern in cylinders 2-4-6 in test No. 7 with the wear pattern of the cylinders with the heavier plate of chrome. It is evident at once that, although the general shape of the pattern is the same, there is a greater amount of wear with the thinner chrome plate in test No. 7. Although the ratio of wear in the two cases is not the same as the thickness of plate, it is

at least three-quarters of that ratio.

The increase in wear due to the thinner coat of chrome was probably due to a longer running time without chrome on the rings. The figures for increase in piston ring side clearance in the groove are not shown in Fig. 1. But they indicate that the use of chrome-plated rings in the top groove tends to reduce the increase in side clearance between the piston ring and the groove for the second ring.

In the lower part of Fig. 1, the increase in piston ring gap is shown for the same tests that are shown in the upper portion of the figure. It will be seen that the wear of the chrome-plated rings is about one-fourth that of the plain iron rings. Also, the rings with the thicker plate wear less than those with the thinner plate, the ratio of wear being about 100 to 85 rather than 100 to 50, the plate thickness ratio.

To obtain information on the importance of clean ventilating air in the crankcase, test No. 12 was run in which the same conditions as those used in test No. 5 were repeated; but instead of using a metal mesh air cleaner to clean the ventilating air through the crankcase, the ventilating air was drawn from the control room after having passed through an absolute filter. In this way the conditions of test No. 5 were repeated, except that absolutely clean air was used for ventilation instead of the air drawn through the usual metal mesh air cleaner.

The results of this comparison are given in Table 5 and show no change in piston-ring gap or bore wear. There are very definite improvements in decrease in bearing weights and in increase in bearing clearances when clean air is drawn into the crankcase. The wear on the bearings is from one-half to one-third when clean air is used. As in both of these tests, no oil filter was on the engine. Note that the petroleum ether insolubles and chloroform insolubles were very high. These figures indicate that a considerable amount of dirt enters through the carburetor and passes the rings, which is not removed when no oil filter is used.

One question which always comes up when a test facility such as the dust tunnel is used is how well can results be repeated? Are the comparisons made accurately enough to justify their use? The dust tunnel has not been in use long enough to give a final answer to these questions, but we have a number of indications at present that are reassuring.

The results given in the lower part of Fig. 1 show how closely the wear on piston rings is reproduced in four tests. The three lower rings in all pistons were of plain cast iron. Although there may have been more chrome present in the oil in some cases, the wear of the rings is quite uniform. Also the wear patterns of the bore shown in the upper part of the figure show very good reproducibility.

To make the reproducibility more quantitative, Table 6 has been prepared. In this table, the wear measurements of comparable tests Nos. 7, 8, 9, and 10 have been tabulated. Since test No. 10, was definitely shorter than the others, an effort was made to bring them more nearly to the same basis by changing the wear figures in proportion to the time of running and to the dust concentration. We are not sure that this procedure is justified, but the results of such assumption are shown in Table 6.

When the brief experience in this equipment is considered, it is believed that the reproducibility of results is quite good. We are confident that they can be improved as our experience develops.

Another question frequently asked is what is the relation between the severe conditions used in a dust tunnel test and operation on paved highways? As yet we do not know the answer to this question, but we are planning to find out.

We realize that the tests to date have been made in a heavy dust concentration and we will soon make tests in a much lighter concentration. We are also running cars on the road with the same engine which has been used in the dust tunnel to furnish more information on this important question.

There are many other unanswered questions. How much dust does it take to ruin an engine? Will an engine wear out more rapidly from dust of large particle size than with dust of small particle size? What size of particle will not injure an engine? We know that some dust is more abrasive than another dust. How can we measure its abrasiveness? Perhaps the tunnel will help answer some of these questions.

Paper on which this article is based is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.

Table 6—Reproducibility of Results with no Protection on Engine and Values Adjusted for Length of Test and Dust Concentration

Test Number	No. 7	No. 8	No. 9	No. 10
Total Test Time, hr	5.6	6.1	6.0	4.5
Average Dust, g per cu ft	0.0131	0.0122	0.0129	0.0110
Equivalent Test Time	5.87	5.95	6.2	3.95
Cylinders Used	1-3-5	1-3-5	None	All
Piston Ring Gap Increase				
Top Compression	0.41	0.37	—	0.45
2nd Compression	0.26	0.22	—	0.29
1st Oil	0.24	0.22	—	0.31
2nd Oil	0.21	0.20	—	0.25
Piston Ring Side Clearance Increase				
1st Compression	0.008	0.008	—	0.009
2nd Compression	0.005	0.005	—	0.006
1st Oil	0.003	0.003	—	0.002
2nd Oil	0.001	0.002	—	0.002
Diametral Increase Bores —Thrust Axis				
Down from top:				
3/8 in.	0.010	0.009	—	0.014
1 1/4 in.	0.008	0.006	—	0.010
3 1/16 in.	0.007	0.004	—	0.006
4 29/32 in.	0.002	0.002	—	0.002
5 1/2 in.	0.001	0.000	—	0.000
Bearing Weight Loss, grams				
Rod	0.38	0.35	0.31	0.32
Rod Cap	0.25	0.20	0.19	0.19
Main	0.35	0.13	0.19	0.21
Main Cap	0.97	0.62	0.57	0.62

How Engine Was Developed for

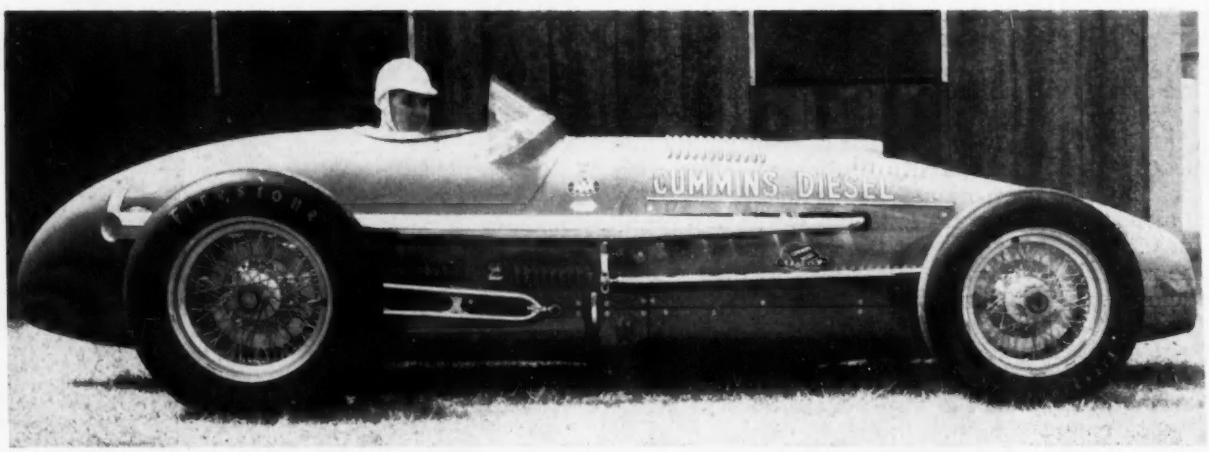
THE Cummins Diesel Special, which broke two world records for diesel cars and set four others (see box), was powered by a modified diesel truck engine. From the less-than-six-month metamorphosis from truck engine emerged a race car powerplant weighing almost half as much as the prototype with more than double its power.

Request to "soup up" the new JBS-600 Cummins commercial diesel engine for a race car entry in the Indianapolis "500" was initiated Dec. 24, 1949. The engine was installed in the race car on May 22, 1950. Between these two dates the research laboratory carried out an intensive development program.

The laboratory had to start with a new, untried truck engine, $4\frac{1}{8}$ in. bore and 5 in. stroke, supercharged for 150 hp at 2500 rpm, which was just released to the production department. There was a dynamometer test engine with a belt driven supercharger so that the amount of supercharging could be varied quickly over a wide range. This engine had been used in determining the air flow and pressure required for the truck engine before the supercharger for that engine was designed. This engine had been pulled at 170 hp briefly.

Production superchargers used on the 275-hp NHS and the 300-hp NHRS were available. In addition, there was some experience gained previously from

Diesel Car Sets Six World Speed Records



On Sept. 10 and 11, at Utah's Bonneville Salt Flats, the Cummins Diesel Special No. 61 proved itself the world's fastest diesel car by breaking international speed records for the mile and the kilometer, and establishing world records for 5 miles, 5 kilometers, 10 miles, and 10 kilometers.

Driven by Jimmy Jackson, shown above, the car covered the mile at 165.23 mph, and the kilometer at 163.83. These topped marks set in 1936 by Capt. George Eyston of 158.87 and 159.10, respectively, for the two distances. The car recorded an official 161.92 mph for 5 miles and 164.25 for 5 kilometers. (In 1934, "Wild Bill" Cummings drove Cummins race car No. 5 at 112.07 and 126.990 mph, respectively, for the two distances; but these were not officially recognized as international records.)

A mark of 148.14 mph was established for the 10-mile run and 147.63 for 10 kilometers.

World's Fastest Diesel Car

EXCERPTS FROM PAPER* BY

J. C. Miller and **C. R. Boll**, Cummins Engine Co., Inc.

Manager, Research & Refinement   Manager, Engine Sales

* Paper "The New Model JBS-600 Cummins Diesel Engine," was presented at SAE National West Coast Meeting, Los Angeles, Aug. 15, 1950.

the test work done immediately after the war on the development of a four-valve head—a design change that was not utilized when it became apparent that the cost per horsepower gained would be excessive.

From the beginning, speed was of the essence.

Quick calculations showed that the valve gear would follow at something over 4000 rpm and the rods would hold together at over 5000 rpm. The engine appeared mechanically sound. Fast slide rule work, using some hypothetical bmep's and rpm's and the liberal use of "if," gave us 225 hp certain and the possibility of 240 hp from the engine.

A review of the qualifying speeds of the last few years, and what we could learn of the horsepower used in those cars, led us to believe that qualifying speeds of 127 mph could be made with our expected horsepower.

Our problem then, was to get the maximum possible horsepower and still hold the engine together. It was necessary to reduce the weight by about 600 lb and to get the resulting engine into a chassis. A quick call to Kurtis-Kraft in California indicated that a car could be made available. A rough outline drawing was sent to them. They quickly determined that, with the addition of 4 in. to the wheelbase of their chassis, the engine could be mounted.

Development of the engine was then started. The date was early January, 1950.

The first step was to set up the standard dynamometer engine with the belt-driven supercharger. The governor and over-speed stop were removed. The stops were removed from the fuel system so that practically unlimited fuel could be poured in. Short runs were made to determine the immediate horsepower gained and the limiting speed without regard for engine life.

The first run gave a maximum horsepower of 187 at 2700 rpm. The engine had 90 hp at 4000 rpm, where it ran for 1½ min before burning up the rod bearings and seizing a piston. While this was a very brief endurance run, it did reassure us that the connecting rods would hold together at this speed, the valve gear would function, and that there was a serious limitation on breathing with the two-valve head.

The engine was rebuilt, giving the pistons more clearance. Revisions to use an old four-valve head saved the earlier experiments. This build-up ran somewhat longer than the first because the pistons no longer seized. However, the rod bearings again failed and the front main bearing failed, due to the belt pull from the supercharger drive. There was, however, an immediate gain of approximately 40 hp with the four-valve head.

The engine was again built up with changes to eliminate these failures. A much more careful job was done in setting up the connecting rod bearings. A continuous groove was put around the center of the rod bearing shell to increase the oil flow and cooling on this bearing.

By this time, our tests had indicated that a rate of air flow approximately equal to that obtained from a standard NHS-600 supercharger, driven at engine speed, was needed. Arrangements were made on this revised engine to drive the blower direct from the front of the crankshaft through a flexible coupling. Not only did this allow the use of a standard blower, but by operating a large blower at engine speed, high blower efficiency was possible. By adding a motor-driven lube pump in parallel with the engine lube pump, the oil flow and pressure were increased.

With this set-up, the engine held together. With the supercharger and valve arrangement at that time, it appeared that the best breathing and horsepower could be had at 3600 rpm, so this speed was used for all development. The program primarily settled down to combustion study. This involved changing piston crown shape, injector spray hole size and angle, compression ratio, and injection timing.

The compression ratio was varied from 11½ to 1, to 15½ to 1. There was little change in power in this range; but the low ratio gave starting difficulty and detonation, while the high ratio increased run-in difficulties and blowby. A compression ratio of

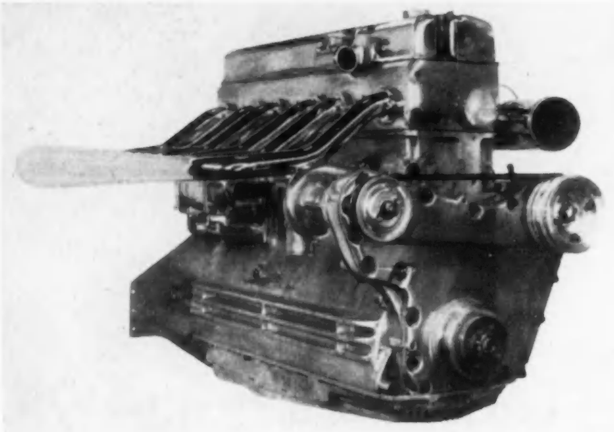


Fig. 1—Manifold side of diesel engine powering the Cummins Diesel Special race car

14¼ to 1 was settled on finally as the best compromise. This compares to a compression ratio of 15 to 1 for the standard truck engine.

Injection timing was varied from 10 deg slow to 12 deg fast compared with the standard truck engine setting. But the standard truck engine setting was finally determined to be best, even at the high speed.

Various fuels were investigated, but for the fuel injection system being used, standard diesel fuel, exactly the same as used on our factory test blocks, was determined best. All subsequent investigation was done utilizing standard diesel fuel.

Our first estimate of 225 hp was quickly passed. By a multitude of increasingly small refinements, a horsepower of 300 at 3600 rpm was finally attained. There was still no idea of the durability of this combination, as the power was taken from hardly more than flash readings.

To get some idea of durability, an endurance run was made in the laboratory. A cycle of operation was figured out which we believed at that time closely approximated the cycle of engine operation on the 2½-mile track at Indianapolis. The engine was set at 300 hp and the speed varied between 2800 rpm and 3600 rpm with the throttle.

An acceleration period of 10 sec and a deceleration period of 20 sec were used. The dynamometer load was kept on at all times. After the engine proved that it could survive this rough treatment for 6 hr, the next phase of the program—reduction of weight —was considered.

The experimental wooden patterns made for cast iron were put in the pattern shop. All fillets and corners filled liberally with wax so that aluminum castings could be taken from this equipment. This was done on the cylinder block and cylinder head and all other parts, such as brackets, pumps, pipes, and covers that could be made in aluminum. Castings were secured, machined, and assembled into an aluminum engine. Neck-down studs and cap screws were utilized extensively to help compensate for the difference between the expansion rate of aluminum and steel.

The aluminum engine ran well and many of the expected mechanical difficulties failed to materialize. There was, however, an immediate loss of power with the aluminum engine of about 10%. Apparently bearing clearances opened up and it was also necessary to increase the oil flow.

At first, the valve and injector push rod clearances varied so much with temperature that the engine could only be started with loose clearances, and readjustments made after the engine had warmed up. The steel push tubes were replaced with solid aluminum tubes and this change corrected the difficulty.

Since the aluminum version of the engine developed no serious mechanical troubles, the combustion study was resumed and the lost horsepower was regained and more. This engine was put through the same 6-hr endurance cycle as its cast-iron predecessor and came through successfully. The engine was now ready for installation in the chassis and actual running on the race track.

The final engine still closely resembles the basic "JS" truck engine from which it was started. The exhaust manifold side of the engine used in the race car, less the supercharger, is shown in Fig. 1.

The major differences between the commercial version of the "JS" and the race car engine are in the use of aluminum cylinder block and heads and the four-valve for better breathing at higher speed. The crankshaft, bearings, connecting rods, cylinder liners, pistons and rings are production parts. The piston crown is modified to improve the combustion at the higher speed.

Fuel System Ups Speed, Output

The valve and injection timing are the same as the standard truck engine, although the spray holes in the injector have been changed for the increased fuel required. The engine has the basic Cummins injection system, with injectors modified to eliminate the necessity for drains and to increase the fuel delivery. Fuel metering is by an experimental system which has been under development in the laboratory for some time and which cannot be discussed at this time. However, this new fuel metering system is definitely responsible for our attaining the high rotative speed and high horsepower output.

Lubricating system is the dry sump type, with separate scavenge and pressure pumps. About 30 gpm of oil are circulated for cooling and lubrication. A standard NHS truck engine supercharger, driven directly from the front end of the crankshaft was used to supply the necessary air. Manifold pressure is carried at approximately 30 in. hg (14.7 psi) at 4000 rpm.

The fuel consumption is surprisingly good, being less than 0.5 lb per bhp per hour at the peak horsepower at 3600 rpm. The weight of the engine is 840 lb without clutch, transmission, radiator, or external oil tank, or approximately 2.47 lb per hp, based on the horsepower actually developed by the engine as entered in the race.

By this time, April 4, 1950, the chassis was available and actual installation was undertaken.

The chassis in which the engine was mounted was made by Kurtis-Kraft, Inc., Los Angeles. It is their standard tubular frame, called the "Championship" chassis, with the wheelbase extended to 104 in. Torsion bar suspension was used on all four wheels. The clutch is a single plate Auburn clutch, modified by the use of heavier springs to carry the high torque of this engine. The transmission is a production Cadillac three-speed gear box. The rear ex-

t ion has been taken off and an adapter added w ch takes a Buick torque tube ball with a Buick u versal joint on the propeller shaft. The rear a e is a Conze axle with quick change gears made in L s Angeles. The ring gear and pinion in this axle w re made by Indiana Gear in Indianapolis. The b kes are Goodyear aircraft-type spot brakes. The d y weight of the chassis is 2125 lb.

The engine, as it was installed in the chassis, had operated for over 70 hr on the dynamometer. Installation indicated modifications of the lubricating oil system were necessary because of road clearance. Need for the separate scavenge pump, pressure pump, and dry sump required considerable work to balance the flow between the scavenge and pressure system, since any accumulation of oil gave a large power loss due to the crank and rods churning oil.

On May 1 the car was put on the track and run for the first time. Jimmy Jackson, a veteran of the Speedway, was at the helm. The car, carrying the numeral 61, was painted green—and promptly dubbed by track followers "The Green Hornet."

Continued Engine Development

Development, however, had not stopped with the taking of the car to the track. A second engine was on the test block in the laboratory and refinements and changes giving small increases in output were being made. This program was to pay big dividends, as we shall see later.

In the meantime, on the Indianapolis track, all the work necessary to fit the car, engine, gear ratios, and driver to the track was being carried out and experience necessitated many changes and additional experimentation. Prior to the race the car was run over 365 practice laps or 925 miles at high speeds.

After the opening day of qualifications, it became apparent that our horizons as to speed were going to have to be elevated. Actually qualification speeds increased from an average of 128.361 in 1949 to 131.045 in 1950 and the "hot" pace necessitated our again revising our horsepower estimates upward.

This is where continued research in the laboratory on the second engine paid off.

Improvements had been made on the second engine in the research laboratory and it was now producing 320 hp at 4000 rpm, with an exhaust smoke level about like a standard, new NHS-600 Cummins Diesel at 275 hp at 2100 rpm. However, due to the air "scoop" to the supercharger on the front of the car, a "ramming" effect was obtained which permitted the engine to run on the track with an absolutely clean exhaust with the test block fuel setting of 320 hp.

Calculations show that this improvement in combustion obtained from the additional air resulted in an additional 20 to 25 hp, or a total engine output of 340 to 345 hp with clean exhaust. This additional horsepower appeared necessary if the Cummins entry was to find a place in the starting field. Therefore, the laboratory engine, with all latest improvements and with the horsepower settings outlined above, was installed in the racer during the week of May 22.

Jimmy Jackson qualified the car on Saturday, May 26, with one day of qualification time remaining, at a speed of 129.208 miles per hour. Qualification consists of a time run of four laps or 10 miles, and the 33 cars with the highest qualifying speed start the race on Memorial Day. One other car subsequently qualified at a lower speed than the Diesel Special, thus, completing a field of 33 on Saturday. This put the Cummins entry 32 in a field of 33, and if two cars were to qualify later at speeds in excess of 129.208, the diesel would be "bumped" from the race.

On Sunday, the final day of qualification, 24 cars were ready and 21 attempted to qualify. However, all failed except one. At the closing of the track Sunday, the green Cummins Diesel Special was assured a place in the starting line-up on May 30.

The actual race is history. The car was driven at a planned race from the start. Jimmy Jackson, our driver, hit a fairly slow pace at the start of about 110 mph. At the end of 10 laps, he began upping his speed to about 117 mph. The car ran smoothly, and Jackson moved up to 119 mph with a few laps at 120 mph. Still running smoothly, Jackson began his next planned upward movement. We hoped for an average of 121-plus. But on the fifty-second lap, Jackson was forced into the pits. Inspection revealed the mounting flange of the torsional vibration damper proper had broken, allowing the outside ring of the damper to hang on the driveshaft to the supercharger.

Without the effect of the damper we considered it too dangerous to continue the race, even if the ring could have been removed. Therefore the car was moved to the garage.

Gains from Racing Engine Venture

While the engine closely resembles the basic "JS" truck engine, it is in no sense a commercial engine in weight, speed, or horsepower. Our horizon on this matter, however, has been increased widely. To summarize:

1. The commercial version of the Model JS-600 engine has already benefited by the research work and will further benefit in the future.
2. The horsepower output of a commercial truck diesel was more than doubled for the race car application through four and one-half months of intensive research and development work.
3. The weight of a commercial truck engine was reduced by 45% for the race car application at the same time the horsepower was more than doubled.
4. Standard commercial fuel oils, presently available, appear to be satisfactory for operation at much higher speeds and higher engine outputs.
5. Little deviation from standard timing and settings appear necessary to operate at higher engine speeds. For example, the standard, commercial timing also proved best at speeds of 4000 rpm. This assured us that no automatic, external injection-timing retarding or advancing mechanism would be necessary for operation at higher speeds.
6. Finally—and most important—we now know how to obtain very acceptable combustion at much higher engine speeds at little or no loss in economy.

Paper on which this article is based is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.

W. J. Davidson

1891-1950

WILLIAM JOSEPH DAVIDSON, president of the Society of Automotive Engineers in 1939, died on September 4 in Goderich, Ontario, his summer home.

He began his engineering career in 1914, the year following his graduation from McGill University with a B.Sc. degree in mechanical engineering, by joining the Cadillac Motor Car Division of General Motors Corp. Except for two two-year interruptions during World Wars I and II, he worked continuously for the Corporation from that time on.

Joining the U. S. Army Motor Transport Corps in 1917, he served in France and later was returned to this country to work on production of military vehicles. He was discharged with the rank of captain. Later he was awarded the Cross of the Legion of Honor by the Government of France for his war service and continued cooperation with French military and civilian engineers.

Mr. Davidson joined the Society in 1926. For two years prior to his term as president, he served as councilor. He had been a member or chairman of the Motor Vehicle, Passenger Car, Riding-Comfort, Military Motor Transport Advisory, and the Constitution Committee, and in 1936 served as chairman of the Engineering Relations Committee. He was the SAE representative on the American Standards Association Sectional Committees on Safety Code and Standards for Inspection of Motor Vehicles.

Upon his return from World War I he was appointed chief engineer of the Canadian Products Division of General Motors with headquarters at Walkerville, Ontario. Born in Montreal, Canada, he became a citizen of the United States in 1924.

After a year as technical service engineer with General Motors of Canada, he was appointed executive secretary of the Corporation's General Technical Committee in Detroit as a member of the staff of Alfred P. Sloan, Jr., then president of the Corporation. This committee planned, supervised the construction of, and at first operated the GM Proving Ground at Milford, Michigan.

In addition to his other duties, Mr. Davidson was named executive secretary to the newly formed New Devices Committee in 1926 and four years later was appointed business director of General Motors Research Laboratory.

In 1934 he became technical director of the Corporation and served under Richard H. Grant, then vice-president in charge of sales.

The year he served as SAE president Mr. Davidson was appointed general sales manager of the new Diesel Engine Division.

In April, 1940, he went to France on aviation problems, and was in that country when it was invaded by Germany. Later he was loaned to the British Purchasing Commission and was recalled by the Corporation late in 1941 to manage the Automotive Ordnance Section. He was director of Engineering Service of the GM War Products Division.

His last assignment, beginning five years ago, was as executive engineer of the Technical Center of General Motors Corp., on the staff of C. L. McCuen, vice-president.

His combined courtliness and warmth made him one of the most popular presidents in the Society's history, and he will be long remembered for his gay stories and anecdotes told in French-Canadian *habitant* dialect.

He is survived by his widow, a daughter and a son, a seaman serving on the U.S.S. Missouri.

Vehicle Designers, Operators From Researches Disclosed

Frank Elliott, chairman of the Sightseeing and Transportation Committee (left) greeting three of the eastern visitors: H. O. Mathews, Transportation and Maintenance Activity meetings chairman; A. H. Fox, SAE Vice-President representing Diesel Engine Activity; and G. M. Shoemaker, one of the speakers

HORIZONS of automotive land transportation were elevated at the SAE National West Coast Meeting, August 14 through 16 at the Biltmore Hotel, Los Angeles. Nearly 500 design and operating engineers heard and saw how more of nature's tightly clenched secrets are being pried loose.

As a result more is now known of the behavior of carbohydrates in engines, how molecules can be made to work with, instead of against designers, why the concept of similar engines can serve us a useful design tool, and under what conditions gas turbines, air cooled gasoline engines, diesel powerplants, and conventional gasoline powerplants have advantages to motor vehicle operators.

"Highways and Horizons," chosen as the subject for the dinner address on Tuesday, was an apt keynote for the three day meeting . . . and "The Road to Engineering Competence" by SAE President James C. Zeder, neatly bundled up the nine-session meeting sponsored by the Diesel Engine, Fuels & Lubricants, Transportation & Maintenance and Truck & Bus Engineering Activites of the Society and its eight West Coast Sections and Groups.

"If this nation is to keep advancing toward the bright horizons ahead, it must do so over a constantly improving highway system," W. T. Gossett, Ford Motor Co. vice-president, told the dinner session.

"In the short span of less than 50 years motor transportation has become a mainstay in our economy. Next to farming, truck driving is the largest single occupation we have, with well over five million drivers earning their living on wheels.

"In all calculations and predictions of the continued expansion of our nation's economy we must face the fact that our present highway plant falls dangerously short of meeting our current needs, much less the probable needs of the future," he said.

Gossett then pointed out that the National System of Interstate Highways, comprising about 38,000 miles or a little more than 1% of our total highway mileage, carries approximately 20% of the nation's highway traffic.

"If as much as $2 billion a year could be devoted to this network, with the remaining highway funds used for the most essential work on other important roadways, we could expect to have our major roads in adequate condition within about five years," he said.

Engineers who develop the greatest reputations

Glean Practical Facts at West Coast Parley

for competency are those who, in addition to having proficiency in some field of engineering, have deliberately and carefully prepared themselves to cope with the human problems of their profession, President Zeder said in a paper prepared especially for engineering students.

"An engineering training is much like a fine set of tools: acquiring them does not automatically make the owner a fine craftsman," he said.

"Knowing the right way to use them and developing the requisite skill and patience are most important.

"Proper understanding of other people will help you most on your road to engineering competence.

"If young engineers study the other fellow's viewpoint, they will gain readier acceptance of his ideas because then they can present their plans in the terms of the benefits to those to whom they are talking or reporting.

"You will be astonished at how your influence and prestige will increase if you make a practice of analyzing and presenting your proposals in terms of the good they will bring to the fellow you are doing business with," he predicted.

Additives Argued

Operators have been milled between the *pro* and the *con* schools of thought about additives to the point where the grindings have become exceedingly small, some at the meeting felt. But one of the 15 speakers at the meeting called attention to the fact that the quality of fuels had been overlooked too frequently in studying engine oils.

This engine engineer pointed out that good lubricating quality, high heat resistance, low carbon formation and non-sludging characteristics are the four major requirements of a satisfactory engine lubricant.

He credited improved heat resistance in lubricating oils to the oil industry, as well as the lower sludge tendencies, through the oxidation inhibitors and detergents.

But chemical additives have already tended to increase the amounts of deposits of oxidized and burned oil residues on the upper part of pistons and in engine combustion chambers, it was stated.

Inferentially either Damon or Pythias had been consulted without the other, because caution was thrown to the winds and the oil industry "started by fortifying the lubricant with ten times as much additive as was used in conventional 'heavy duty' oils."

Then "deposits disappeared as if by magic, wear life was doubled, engine parts remained cleaner than ever before, and even the lowly oil filter profited by the 'Super Oil.' "

Because the filter was unable to retain much of the loosened contamination, its apparent change periods increased tremendously, the author recalled.

But in the meantime a marked improvement has taken place in the quality of fuels. The remarkable come-back in fuel quality, he said, has eliminated the need for lubricants with extremely high additive content.

"The tremendous cleansing power of high additive content in engine oils is dearly paid for by their almost complete lack of filterability," he said.

"During the past 15 years the petroleum industry has made tremendous strides towards the supply of motor lubricants combining improved lubricity with higher heat resistance and lower sludging tendency.

"But even the best lubricants cannot be expected to be a substitute for good fuel," he concluded.

Molecular structures have now been used successfully to bond two unlike metals, it was revealed at the meeting. How this recent manufacturing process has been used to multiply piston and cylinder life was detailed by one of the developers of the method, and still another engineer explained some research work on piston cooling and ring groove wear, calculated to decrease maintenance costs by another approach.

Thousands of bimetallic pistons are in use today, the former reported. Many are in heavy duty diesel engine service. Examination of one, which had run 147,000 miles, showed no positive evidence of wear—even in the top ring groove which was machined in the high nickel cast iron ring groove insert in the aluminum alloy piston.

The bond of the ferrous and aluminum components of the piston is a molecular one, it was pointed out. Because the two metals have been cast without

General Chairman J. W. Sinclair (left) introducing Paul P. Olson, chairman of the Tuesday morning session

failure of the bond in the casting, and the nickel alloy ring groove insert has a lesser coefficient of expansion than the aluminum, the heating cycles during the piston's life puts the insert in compression. This favors the bond.

It was reported that the theory works in practice, permitting engineers to take the advantage of the light weight, low inertia aluminum piston head and skirt but multiplying ring groove life with the tough nickel alloy insert.

Another speaker reported a new study of piston cooling. This project was undertaken to throw new light on the effect of piston cooling, particularly in respect to ring groove wear. A special aluminum alloy piston was used.

"Any reduction in temperature through oil cooling will greatly improve the physical properties of the piston material, will reduce ring and ring groove wear, and decrease the tendency of rings to stick due to high operating temperatures," he reported.

Laws of Similitude

The laws of similitude, long used by mathematicians, have been put to work in an analytical method to study internal combustion engines, a speaker from the east told the West Coast Meeting.

Assuming that engines of any group under study are similar, irrespective of their sizes, analyses can be made as readily as those of thermodynamics of engine cycles, bearing loads, balance, and vibration, he explained.

The conception of similar engines can be extended to cover similar cylinders on engines which have different cylinder arrangements if each cylinder is considered as an independent unit. In this case, he said, the similitude would not extend beyond the cylinder assembly, valves, and pistons.

A striking example was the comparison of a 8000 hp, 29 in. bore, two-stoke diesel engine, with a half-inch bore two-stroke model airplane engine without fuel injection.

Such factors as bmep, piston speed, hp per sq in., and weight per cu in. of displacement showed up roughly the same, although the horsepower per cylinder of the big powerplant was 710 against 0.136 with piston displacements of 26,500 cu in. in the large engine and 0.10 cu in. for the small one.

"Mean effective pressure and piston speed are universal measures which are nearly independent of engine size, but very dependent upon engine design," he pointed out.

"Thus the theory of engine similitude affords an extremely powerful tool which appears to have been neglected.

"With appropriate modifications this theory can profitably be applied to other types of machinery," he said.

Truck Gas Turbines

Engineers found out much more than had been generally known about the performance of gas turbine powerplants for land vehicles when another author took his audience behind the scenes of one of today's most dramatic automotive development projects.

Although the Boeing gas turbine for trucks had run only 150 hours and 2200 road miles, the author who described the powerplant and the tests to date concluded that there is a definite field for this type of prime mover in highway transportation.

The model 502 engine was installed in a Kenworth truck. At the front of the gas turbine, as installed in the test truck, is an air intake silencer. The turbine drives through a coupling into a marine reverse gear, 1:1 both in forward and reverse.

Behind this is a seven speed and reverse planetary transmission, working into a three speed auxiliary gear box. Behind that is the conventional rear axle.

The reverse marine gear produces conventional rotation in the power train required, because the turbine output shaft works in the opposite direction. It also enables using power to retard the speed of the vehicle on down grades.

Work done to date by Boeing engineers, the speaker said, indicates that higher efficiencies in fuel use are in the offing. To obtain maximum overall efficiency, he said, considerable special design problems must be worked out on the vehicle to be powered by the commercially acceptable gas turbine of the future.

Diesel vs. Gasoline Trucks

Wednesday's symposium on Diesel versus Gasoline Powered trucks proved to be so enlightening that the four authors served as a panel of experts for the evening session that day, and answered knotty questions from the floor.

Perhaps never before has the question of which of the basically different truck powerplants to use had as objective a discussion as at this meeting. Academic discourses of the past few years were forgotten in view of a mass of operation data that has been accumulated and disclosed at the meeting.

For example, the case of the diesel engine was made with more test data than had been accumulated until now, and the proponent of gasoline powerplants was equally as factual.

Emphasis by two of the four authors was laid on

economic considerations and the crux of the decision as to which the operator should buy was said to be his proposed use of the vehicle.

A highlight of the symposium was a careful presentation of the aircooled engine's prospects in the highway transport field. The paper was partly a description of the air-cooled Army tank engine of World War II, which resulted from a heroic development project initiated in 1932 in the face of abandonment of this type of powerplant in this country and in Great Britain in favor of water-cooled engines.

A combination of vision and ceaseless effort produced the prototype of 100,000 air-cooled tank engines during World War II, and interchangeability found a new and a brilliant example in that achievement.

Principal requirements of a military engine and a commercial truck prime mover are, generally, identical. But low weight, small size for the required power, and low maintenance costs are sufficiently important to truck operators to be among the deciding factors.

The author of the aircooled paper made a strong case on these factors, and showed that the lesser considerations were about equal. His tabulations and discussions included diesel engines, which was most helpful in giving fleet engineers a rounded picture of what may be expected in powerplants of these types today.

An exhaustive research which evaluated fuels for high compression engines disclosed that the problem of satisfying high compression engines is basically no different from that of satisfying present production engines, it was disclosed at another session.

It was shown that the GMC Research high compression test engine used became milder at all engine speeds as compression ratios were increased.

This means, it was claimed, that future high compression engine designs will tend to extract the full benefit of future gasolines in view of the present refining trends toward producing gasolines of greater sensitivity.

As speed was increased the engine became more severe at any compression ratio. Data produced indicated that the diisobutyleneheptane blends may be useful to engine manufacturers in studying factors affecting engine severity.

Under the general chairmanship of **J. W. Sinclair**, the following served as chairmen of the eight technical sessions of the SAE National West Coast Meeting: **W. D. Stewart, F. D. Chapman, H. N. Taylor, Paul P. Olson, F. G. Backman, L. P. Johnson, Reagan C. Stunkel,** and **H. L. Stone.**

This report is based on discussions and 13 papers . . . "Gas Turbine Propulsion for Ground Vehicles?," by **W. M. Brown**, Kenworth Motor Truck Corp.; "Correlation and Presentation of Diesel Engine Performance Data," by **C. Fayette Taylor**, Massachusetts Institute of Technology; "Piston Cooling and Ring Groove Wear," by **Frank Jardine**, Aluminum Co. of America; "Bi-Metallic Pistons," by **C. E. Stevens, Jr.**, Chicago Railway Equipment Co.; "The Evaluation of Motor Fuels for High Compression Engines," **M. M. Roensch**, Ethyl Corp.; "High Additive Motor Oils vs. Fuel Quality and Engine Life," by **H. M. Gadebusch**, Detroit Diesel Engine Division, GMC; "High Additive Motor Oils vs. Engine Design and Operating Conditions," by **C. W. Georgi**, Quaker State Oil Refining Corp.; "Accessibility," by **Robert Cass**, White Motor Co.; "The New Model JBS-600 Cummins Diesel Engine," **J. C. Miller** and **C. R. Boll**, Cummins Engine Co.; Symposium on Diesel vs. Gasoline Powered Trucks—"Basic Reasons for Preponderance of Gasoline Powered Trucks," by **L. L. Bower**, Waukesha Motor Co.; "Recent Advances in Gasoline Engines for Transport," by **Robert Insely**, Continental Aviation & Engineering Corp.; "Fundamental Advantages of Diesel Engines for Trucks," by **F. G. Shoemaker**, Detroit Diesel Engine Division, GMC; "Fuel Economy Depends on Proper Engine Application," by **C. R. Boll**, Cummins Engine Co.; "The Road to Engineering Competence," by **J. C. Zeder**, Chrysler Corp. . . . **All of these papers will appear in abridged or digest form in forthcoming issues of SAE Journal, and those approved by Readers Committees will be printed in SAE Quarterly Transactions.**

Power Steering for Earthmovers

Continued from Page 45

der, causing a left-hand steering movement. The oil in the top of the left-hand cylinder and the bottom of the right-hand cylinder then returns under low pressure back into the reservoir in the steering head, and from there into the reservoir tank in the side of the frame. When the steering wheel is turned to the right, this procedure is reversed.

This steering head is a progressive type of steer in that if the operator wants to continue turning, he must keep turning the steering wheel. If he should stop turning the wheel, the spring-loaded plunger automatically centers itself, locking the steering system in that position.

If, for any reason excessive pressure is encountered in steering, the relief valve opens and bypasses the oil into the reservoir chamber. A safety check valve is provided in the center port which closes if excessive kickback pressure is encountered. This prevents damage to pump and valve mechanism.

This steering also has a safety feature in the event of power pump failure. The operator, by manually rotating the steering wheel, will cause the metering gears to become the power pump, drawing oil through the check valve from the reservoirs and forcing it into the steering cylinders. If the hydraulic system should fail completely, steering still can be controlled by operating an air valve located on the side of the steering head.

This air valve controls the individual braking of the driving wheels, and this method of steer can be utilized until adequate repairs are made to the hydraulic system.

Paper on which this article is based is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members. 50¢ to nonmembers.

Why Transonic Speeds Bring

AS an airplane increases speed through the transonic speed region, aerodynamic changes take place which markedly affect airplane stability and control:

- Shock waves form on the airfoils after the local speed of sound is attained. Pressure distribution over the airfoil rearranges. The wing aerodynamic center—normally about 25% of the chord distance from leading edge to trailing edge in subsonic flight—moves rearward to about the midpoint in true supersonic flight.
- Flap-type controls lose some of their effectiveness for two reasons: When local air speed becomes sonic, the effect of a flap cannot be transmitted forward of the hinge line. Deflection into the low-energy, highly turbulent boundary layer has little useful aerodynamic effect on the surface. It merely causes buffeting.

(This boundary layer begins at aft chord positions and is thin at first. As the aircraft increases speed, the boundary layer begins farther forward on the airfoil and its depth increases. Boundary layer depth and extent are also a function of angle of attack at supercritical speeds. With increasing angle, boundary-layer depth and extent increase on the upper surface of the airfoil and decrease on the lower, as Fig. 1 shows. Increases in angle of attack increase lift, but the increment is smaller the higher the airfoil speed. Therefore, to maintain constant lift coefficient as airplane speed is increased, angle of attack of the airfoil, such as the horizontal tail, must be increased.)

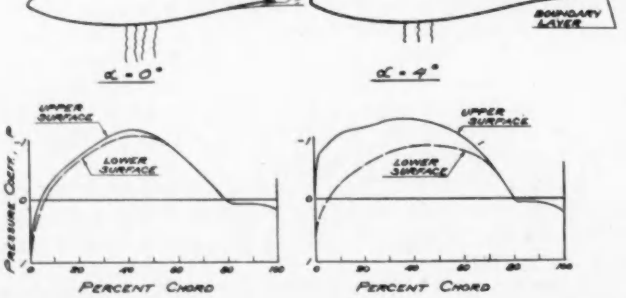


Fig. 1—Effect of angle of attack on characteristics of NACA 65, 3-019 airfoil at supercritical speed of M = 0.678

- Flexibility of the airplane acts to change the local spanwise lift-versus-angle of attack relations through a combination of twisting and bending deformation. Fuselage bending generally tends to reduce tail surface effectiveness.

Of course, flexibility is not a strictly transonic problem, but it is more important in the transonic region—especially at low altitudes, where dynamic pressures are high.

- Average wing downwash angle at the horizontal tail changes.

Indications are that compressibility effects on the stabilizer-elevator combination are responsible for most of the apparent trim changes of high-speed aircraft flying in the transonic range.

Republic's Experience

Republic's experience with a high-speed jet-propelled fighter illustrates these points.

At high indicated speeds and moderate to large airplane load factors, the elevator stick force on the original airplane tended to reduce with increases in load factor with tip tanks installed (Fig. 2). As a result, the pilot could inadvertently exceed the airplane's design load factor if he did not use extreme care at high indicated speeds.

Wing twist was measured in flight with and without external tip tanks installed, to determine wing divergence speeds and also to learn more about the characteristics of the wing. The moments caused by the tanks on the wing were determined by a strain-gage method. Wing divergence was found to be favorably high and thus not a factor in the problem at hand.

Airplane Trim Changes

EXCERPTS FROM PAPER* BY

Robert B. Liddell, Aerodynamicist, Republic Aviation Corp.

* Paper "Some High-Speed Experience with Fighter-Type Airplanes" was presented at IAS-SAE Metropolitan Section High-Speed Flight Symposium, New York, March 16, 1950.

From these data and wind-tunnel data, it was concluded that the reduction in stick force gradients with load factor was due to both the increase in wing-lift-curve slope due to elastic effects and the increase due to increase in Mach number. These effects are greatly aggravated by the tip tanks. Thus additional wing lift causes additional load factor without any additional elevator deflection or stick force. Also it is probable that the variation in the elevator-hinge-moment parameters and basic airplane stability are contributing factors at the highest load factors.

(It is interesting to note the effect of the fuel load in the tip tanks in Fig. 2. While it causes a favorably steeper force gradient at load factors, the force reversal occurs earlier.)

Two flattened wing tip tanks were designed to reduce the unstable pitching or twisting moment of the tank about in half. These tanks were of the same length and volume as the original round cross-section tanks but had an elliptical cross-section with the major axis vertical. The tank depth was about twice the width. A few qualitative accelerated maneuvers indicated to the pilot that the improvements in stick-force gradients were small. The tests were terminated because of the very detrimental and erratic effect of the flattened tanks on lateral stability. In turns, the airplane would tend to roll out inadvertently at high load factors and start to turn in the opposite direction. Also the tanks tended to buffet at all speeds and reduce the apparent aileron effectiveness.

Finally a tip tank fin was incorporated on the outboard side of each external tank. This cured the troubles encountered in maneuvering. The fin permitted the airplane to be maneuvered up to its design capabilities with stick forces increasing properly with load factor. The tip tank fin area required to do this job has a disproportionately small size in comparison to the size of the tank. This is because the very favorable vortex flow field around the tank on the wing tip caused the fin effectiveness to be two or three times its normal free air value. These fins moved the center of pressure of the tank-fin combination close to the wing's elastic axis and thus eliminated the unstable twisting moment of the tank.

Trim changes on this same jet fighter airplane occur shortly after 0.82 Mach number. These effects are manifested in a strong tucking-up tendency, preceded by a small tucking under, as the speed is increased beyond this Mach number at zero stabilizer setting. While the trim change occurs at all altitudes, it is particularly noticeable at the lowest altitudes because of the higher stick forces required to maintain level flight. At about this trim-change Mach number, a certain amount of airplane buffet starts in level flight and becomes more severe as the Mach number is increased.

In an attempt to discover and remedy these limiting features, an extensive flight test program was undertaken. A larger wing-fuselage fillet located at the trailing edge of the wing juncture was tried, because it was thought that some local sepa-

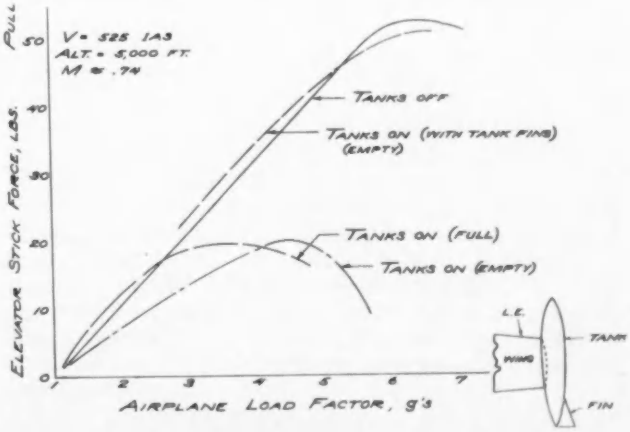


Fig. 2—Effect of tip tanks, tip tank fuel, and tip tank fins on maneuvering stick force gradients

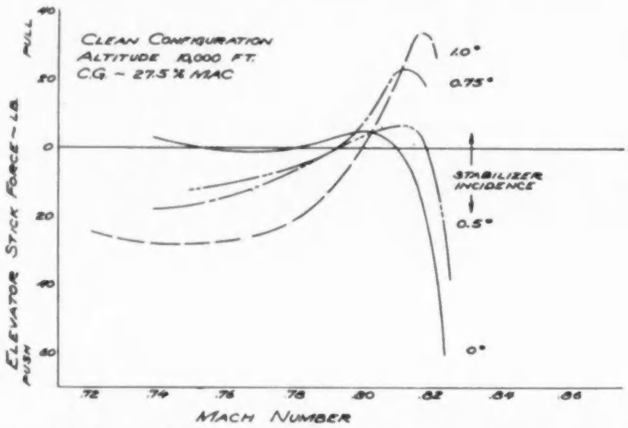


Fig. 3—Compressibility trim changes with various stabilizer settings

ration might be causing an airflow change at the tail. This did not appear to be the case, as the airplane reacted much the same with this large fillet installed.

Separation over the canopy was also suspected, so high-speed pressure distributions over the canopy were recorded. An analysis of the data indicated that no separation caused by Mach number was present. The canopy's critical Mach number was found to be favorably high, and the agreement with low-speed wind-tunnel distribution was found to be very good.

Recent NACA research at Ames Laboratory indicated that effectively changing the average wing camber by rigging upward the flaps or ailerons or both would change the trim characteristics in the transonic region. This scheme was tried with various combinations of flaps and ailerons rigged up and down, but the airplane high-speed trim remained essentially unchanged. It was believed by some that the relatively large wing trailing-edge angle could cause a trim change at the wing's critical speed and also lead to early buffet. This theory was somewhat substantiated in small-scale wind-tunnel tests.

In order to test out this theory in flight, modified ailerons and flaps were built and installed on the airplane. The modifications consisted of extending the wing chord 3% and accentuating the trailing-edge cusp from the 80% chord position. Flight tests showed the airplane's buffet boundary remained essentially the same. The elevator stick forces measured in the pitch-up condition were also about the same.

One of the most promising contrivances to delay the transonic trim change appeared to be small spoilers on the wing leading edge next to the fuselage. These triangular cross-section spoilers were about 20 in. long and ¼ in. high. The spoilers tended to delay the airplane trim changes to a higher Mach number. Wing pressure distributions on the wing section behind the spoiler indicated that a partial reason lay in the reduction of lower-surface critical-pressure peaks by the spoilers. With the wing spoiler installed, the elevator required for trim indicated that this modification acted much the same as setting the stabilizer nose up 1 deg. The airplane buffet was not affected by the addition of these leading-edge spoilers, but the stall warning buffet and stalling speeds were somewhat increased.

Various positive stabilizer settings were tried on the airplane in order to analyze more fully the causes of transonic trim changes. The results in terms of elevator stick force required are shown in Fig. 3. This figure indicates that the slight initial

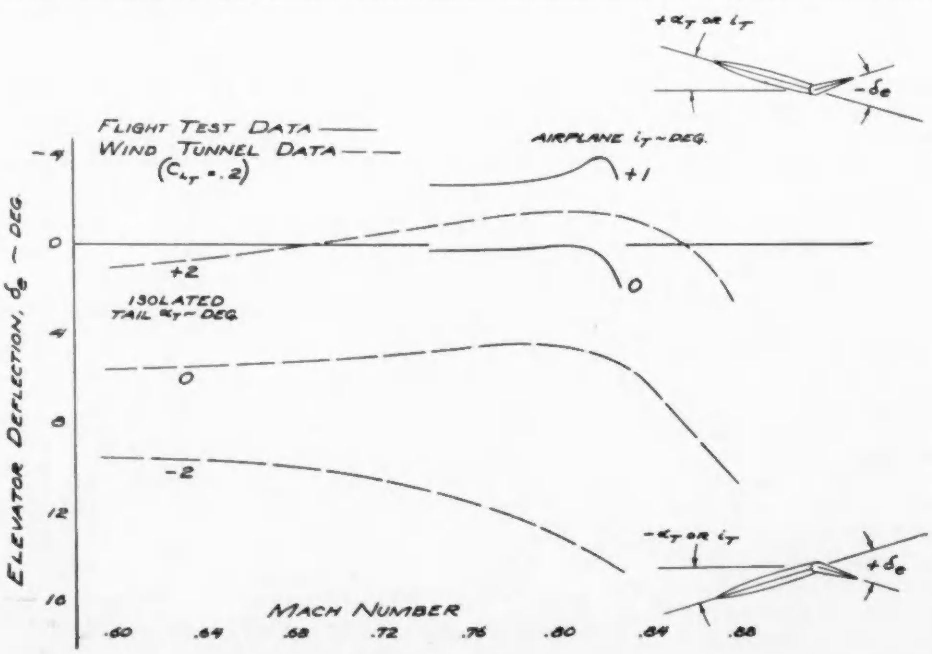


Fig. 4—Comparison of airplane compressibility trim changes with isolated tail

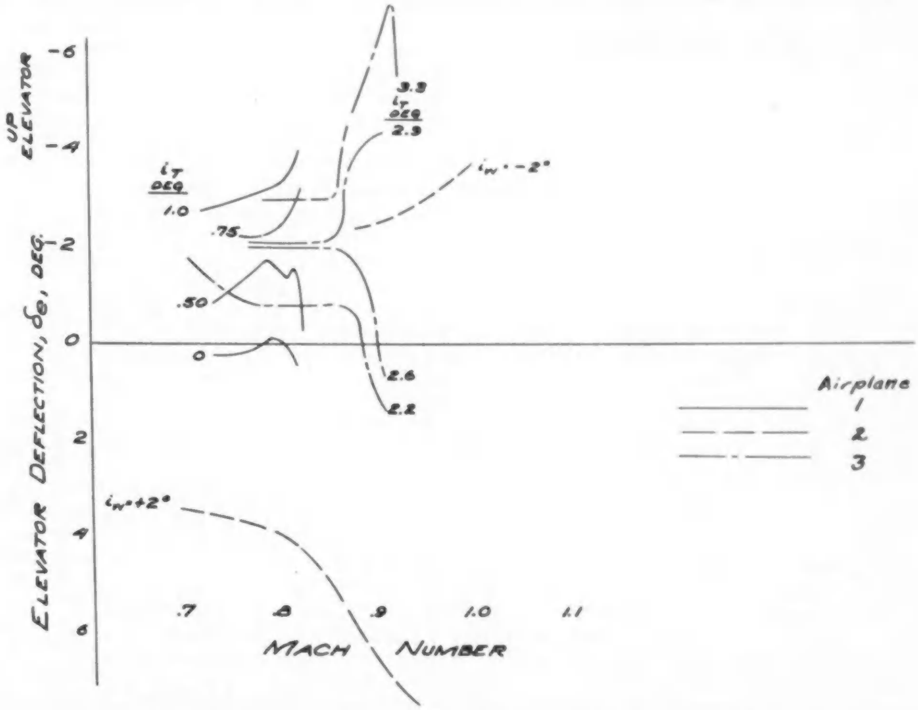


Fig. 5—Effect of stabilizer setting on airplane trim changes in level flight

pitch-down that occurs before the pitch-up is greatly accentuated.

The corresponding elevator deflections required for various stabilizer settings are shown in Fig. 4. Data are also superimposed on this figure of the isolated tail characteristics of another fighter aircraft. This tail is also unswept with similar planform, section, and thickness. These isolated tail data are for a constant small positive tail lift coefficient similar to that required to trim the airplane at the trim-change speeds.

Trim Changes Blamed on Tail

The remarkable similarity in shape and trend of these curves would lead one to suspect that the tail's own inherent characteristics are a major contributing factor to the apparent trim changes that occur near critical speed. The shapes of these curves can be explained as follows: Examination of the top curve of the most positive tail angle of attack indicates that in order to keep a small positive lift coefficient on the tail before critical speed, increasing negative or up elevator deflections are required. This is because increases in Mach number increase the slope of the tail lift curve up until critical, as mentioned previously. After airfoil critical is attained, the elevator deflection required to keep this same lift coefficient must reverse its trend and move in a down direction. This is because lift on the main airfoil is lost after airfoil critical is attained. The small upper sketch in Fig. 4 illustrates this condition.

Examination of the lower curve of a zero or negative tail angle of attack condition indicates that a positive or down elevator deflection is required to keep positive upward lift on the complete tail. As the speed is increased to critical, more and more down elevator is required, because the down lift on the main surface caused by the airfoil negative angle of attack increases with Mach number. After critical speed is attained, the elevator deflection necessary increases to further down deflections because the lower-surface separation due to shock on this surface causes the down elevator to be less effective.

While shifts in wing zero lift, changes in wing stability and downwash do tend to cause trim changes, it is believed that a good majority of the apparent trim changes of the airplane are caused by the compressibility effects on the stabilizer-elevator combination. Calculations have tended to substantiate the relative magnitude of all these effects.

To support this contention, the elevator deflections required for trim on a number of other high-speed airplanes are shown in Fig. 5. They show the same tendencies. Thus, depending upon what setting of the stabilizer was used, the pilot could report a stable or unstable variation in elevator deflection with speed at certain Mach numbers. While this effect is present also on sweptback tail surfaces, data indicate the changes are very much more gradual and occur at greater Mach numbers. This can be seen from the data on Airplane 2 in Fig. 5. This airplane has sweptback wing and tail surfaces. Airplane 1 and Airplane 3 are straight wing and tail airplanes.

(This paper is available in full in multilithographed form from SAE Special Publications Department. Price 25¢ to members, 50¢ to nonmembers.)

Problems in Design Of Large Aerial Cameras

Based on paper by

DUNCAN E. MACDONALD

Boston University

IF large aerial cameras are to operate effectively at high altitudes, the aircraft designer must cooperate with the camera expert in determining the placement and mounting of the camera. In particular, the selection of the photographic compartment of the plane should not be based upon available space but rather should be planned with due consideration of the vibration and air turbulence characteristics of the plane.

Image Motion

With present-day standard equipment, image motion is probably the worst offender as far as deterioration of image quality under flight conditions is concerned. Rolling, pitching, and yawing of the aircraft and the higher frequency vibrations introduced by the engines, propellers, and beat frequencies all contribute to this deterioration.

One method of evaluating the effects of accidental camera motions is to fly the aircraft in a straight, level pass over flashing lights, with the camera shutter locked open. If the camera is held straight and level, a series of dots will appear equally spaced in a straight line across the film. If, however, the aircraft rolls, the dots are displaced from the straight line; if the aircraft pitches, the spacing of the dots is alternately lengthened and compressed; if the aircraft yaws, two simultaneous dot trails indicate this effect, in that the normal at the line connecting two points simultaneous in time will rotate about the optical axis of the camera.

This method of testing enables us to observe all the frequencies of vibration that produce image motion up to the frequency limit of the flashing light. With rotating mirrors it is possible to impress as many as 20,000 flashes per sec.

One finds that the most serious factors are in the zone of frequencies less than 18 cycles per sec and that the overall aircraft motions are of about the same order of magnitude as the high frequency vibrations in their contributions to the deterioration of image quality.

An ideal case for the camera would be a gyrostabilized platform to support the antivibration mount that holds the camera.

It would also be helpful if small cameras would be mounted in batteries instead of individually in available sections of the aircraft, as is now done.

If it were possible to provide vibration-insulated sections of platforms in aircraft designed for photographic purposes, where the cameras would be mounted as a group in this section, it would provide a great advance in the field of aerial reconnaissance.

Air Turbulence

Turbulence is one of the limitations that the earth's atmosphere introduces on the performance of large aerial cameras, for as the size of the lens goes up, both in linear aperture and in focal length, the effects of turbulent air become more severe.

Although some of the air turbulence is caused by microweather conditions—and is thus outside the domain of the aircraft designer—a large part of it is due to air vortices under the photographic window of the aircraft. Thus, if the aircraft designer works with the photographer in placing the photographic section of the aircraft over areas of minimum air turbulence, an image of much better quality can be obtained. (Paper, "Design Considerations for Large Aerial Cameras," was presented at the SAE National Aeronautic Meeting, New York, April 19, 1950. This paper is available in multilithographed form from the SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

Sees No Radical Change in Car Design

Based on paper by

D. S. CARIS

General Motors Corp.

SUBSTANTIAL improvement can be made in the fuel-engine relationship by redesigning engines to gain mechanical octane numbers, particularly in the matter of combustion chamber design and turbulence. This gain will be in addition to those made by increasing compression ratios which can be lifted to 12:1 to improve fuel economy from 20 to 30%.

Claims made that removal of red hot exhaust valves would result in 20 to 30 mechanical octane numbers and permit operation at a 12:1 ratio on 80 octane gasoline are groundless. Tests with sodium cooled valves showed no gain whatsoever, and further tests in which the valve was cooled with tap water to the temperature of the combustion chamber walls, clinched the point.

Automatic transmissions will be used more widely even though they are complicated and expensive to build. It's doubtful if they can ever be made as cheaply as the hand operated gear-shift, but customers want them and are willing to pay the price.

(Paper "Future Trends in Automobiles," was presented at SAE Detroit Section, Mid-Michigan Division, Owosso, March 27, 1950. It is available in full multilithographed form from SAE Special Publications Department. Price: 25¢ each to members, 50¢ to non-members.)

Earthmover Blades Field Tested for Wear

Based on paper by

R. F. BOURNE

Colorado Fuel & Iron Corp.

GRADER blades and other cutting edges used on modern road graders, crawler and wheel type bulldozers and scrapers, are subjected to terrific torque, impact, and abrasion, but attempts to increase abrasion resistance by increasing the carbon content of the steel used have been generally disastrous. Hardness is not the sole criterion governing abrasion resistance. A high carbon content induces brittleness and lower impact strength.

Alloys, heat treating, and hard surfacing have been used in the expectation of lengthening blade life, but they have brought little or no improvement in wear.

To evaluate attempts to increase blade life, controlled field tests have been run on motor grader blades, using double beveled curved sections in the "as is" condition, blades with field applied hard surfacing of various types, 14% manganese blades and shot blasted blades. Wear was determined by comparing loss of blade weights after test runs and costs were figured on a dollar per hour operation basis after taking into consideration blade cost.

In all tests, the field hard surfaced blades showed a greater wear than standard blades. However, it is suggested that blades hard surfaced under controlled factory conditions where they could be pre-heated, then stress-relieved after hard surfacing, might show up better. The 14% manganese blades showed a much higher cost than standard blades, but if work hardened by being run in rocky soils wear resistance would increase. Therefore, under certain soil conditions these blades might give a longer life, justifying their higher cost. Shot blasted blades were also more costly than

standard blades.

Tests were also run on dozer blades, using an "as rolled" blade for basis of comparison. In hard and abrasive soil the 14% manganese blade showed a 2 cents per hour saving over the standard blade. A blade with flame hardened edges showed a rate of wear identical with the standard blade, but the higher original cost makes its operating cost higher. Factory hard surfaced blades showed a great superiority over non-hard faced and field hard faced blades. They also have self-sharpening characteristics superior to field faced or non-faced blades.

The best cutting edges obtainable for average work, when geographical location and special conditions are excepted, are: Grader blades made of plain carbon steel of AISI-C-1085 analysis, ⅝" thick; bulldozer blades of AISI-C-1060 or 1065, plain carbon steel. If rocks and boulders are to be handled, then cast manganese edges are best. Scraper cutting edges give best performance if heat-treated and hard surfaced by the factory.

The paper also discusses some past tests of blades. (Paper "Grader Blades and Other Cutting Edges, Requirements and Specifications," was presented at Earthmoving Industry Conference, SAE Central Illinois Section, Peoria, April 11-12, 1950. It is available in multilithographed form from SAE Special Publications Department. Price 25¢ to members, 50¢ to non-members.)

Suggests Ways to Handle Jets on Ground

Based on paper by

E. C. TAYLOR
American Airlines, Inc.
and
R. E. SMALL
General Electric Co.

This paper was presented in the form of questions and answers:

Question: What will be the method and length of time for starting turbojet engines promptly, including fire guard coverage?

Answer: Equipment generally used has been a 24 volt, d.c. electric starter and idling speed is reached from 30 to 60 seconds after starter circuit is energized. Work is being done on 120 volt, d.c. starters and this type may be used on 1955 transports, but it is too early to say as future selection will depend not only on power requirements, but on the economy of the system. Existing fire extinguishing equipment will be adequate.

Question: Will jets be maneuverable under engine power in and around close tolerance areas at passenger terminals?

Answer: No.

Question: How will they be moved from loading position to the end of the runway for the take-off?

Answer: One method would be to tow the plane a short distance from the loading platform, start all engines from ground-power supply, and taxi to the end of the runway with all engines idling at lowest possible rpm until take-off clearance is given. A second method would use only enough engines for taxiing and start the remaining engines only after clearance. A third method calls for towing to the end of the runway, while a fourth would involve towing a short distance from platform, then starting of all engines for higher speed taxiing to end of runway and full power engine run-up as the plane reaches the runway. Choice will depend on the best compromise considering fuel burn-off, ground maneuverability, and mixed operations.

Question: What will be the procedure at the end of the runway during an air-traffic hold?

Answer: All engines but one might be shut down, to be restarted from power generated by operating engine when the take-off clearance is given.

Question: How much separation will be needed between aircraft during ground movement under engine power?

Answer: The distance probably will be between 100 and 150 ft. During engine run-up, craft will be turned so that jet blast will not be discharged in the direction of planes lined up behind.

Question: Will areas have to be constructed for engine run-up checks which are part of normal operation?

Answer: Expensive test areas are not required. Where propeller craft are now run-up in front of a wind screen, it may be necessary to construct an inclined barrier to deflect blast.

Question: Will the jet be conducive to single engine operation as now used by certain short-haul operators with 2 minute ground time?

Answer: Yes, where transport need not be taxied into a congested area. Dissipation of jet wake while taxiing will require an unobstructed area 100 ft. in length behind jet nozzles.

(Paper "Ground Operational Problems with Jet-Powered Aircraft," was presented at SAE National Aeronautic Spring Meeting, New York, April 17-20, 1950. It is available in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to non-members.)

Higher Earnings Seen For Jet Air Transport

Based on paper by

J. A. MORLEY
A. V. Roe Canada, Ltd

TURBOJET transports can earn 65% higher profits for airlines, despite their higher direct operating costs.

First comeback to any economic claims for the turbojet over propeller-driven aircraft is the jet's high fuel consumption. Admittedly, jet engines have a notorious thirst for fuel. But secret of its economy lies in its speed.

The jet-powered airplane flies more miles per day. It can earn additional revenue to more than offset higher fuel costs. Table 1 compares, expenses,

Table 1—Comparing Earning Power of Air Transports Powered by Piston and Jet Engines

Description	Aircraft A (4 Piston Engines)	Aircraft B (4 Jet Engines)
Block distance	500	500
Block speed, mph	240	320
Payload, tons	6.5	6.5
Direct operating cost/hr	$190.00	$275.00
Overhead/hr	190.00	190.00
Total operating cost/hr	$380.00	$465.00
Direct operating cost/aircraft mile	$ 0.79	$ 0.86
Total operating cost/aircraft mile	1.58	1.45
Direct operating cost/ton mile	12.1¢	13.2¢
Total operating cost/ton mile	24.3¢	22.4¢
Revenue earned in 10 hr flying, at 65% load factor and at 50¢/ton mile	$5060	$6750
Expenses for 10 hr flying	3800	4650
Profit	$1260	$2100

income, and profits of aircraft powered by both jet and piston engines. Although the conventional airplane incurs 10% lower direct operating costs, the jet transport realizes 65% greater profits.

Other savings with turbojets also help offset higher fuel costs. Jet engines require less maintenance because they haven't the complications of propellers, engine superchargers, propeller controls, oil coolers, radiators, and cowl flaps.

The jet's high speed makes it possible to run a given operation with fewer aircraft. This reduces total airframe and engine maintenance costs, depreciation, hangar space, ground handling equipment, spares, and crew expenses.

Intangible items, tough to assess in dollars and cents, also favor the jet transport. Riding comfort for the passenger and handling ease for the pilot are just a few. (Paper "An Economic Evaluation of the Avro Canada Jetliner," was presented at SAE New England Section, Boston, May 2, 1950. This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

25 Years Ago

Facts and Opinions from SAE Journal of October, 1925

Lillian M. ("cheaper-by-the-dozen") Gilbreth of Gilbreth, Inc., spoke at the SAE National Production Meeting. Said that all training done in industry is really training for production, whether the worker being trained holds a job in production, in selling, accounting, or elsewhere.

■

With the growing need for night-driving, highly satisfactory and also uniform headlamp regulations are requisite. Anything else will militate against normal development in the use of automobiles.—Chronicle & Comment.

■

P. E. Flandin, president of the Aero Club of France, and Louis Breguet, prominent French airplane manufacturer, attended the SAE National Aeronautic Meeting at the Hotel Astor, New York, on Oct. 7.

■

In the 12 months up to Aug. 31, 902 applications for membership were received. The rate at which applications have been received this year has increased 30% as compared to last year.

■

K. T. Keller, vice-president and general manager of General Motors of Canada, Ltd., gave a most convincing and helpful message to the 200 members and guests whom President H. L. Horning welcomed at the production dinner in Cleveland on Sept. 15. A great many cost figures, he said, are not understandable, but the executive can secure helpful figures if he knows and will state what he wants.

■

Aluminum alloys are being used to great advantage in the automotive industry. . . . To meet the need for a small number of standard alloys, aluminum-alloy specifications that possess the best all-round physical characteristics are given in the SAE Handbook.—Automotive Research.

■

Recommended revision for an SAE Standard: "Standard oversize pistons for passenger car, marine and airplane internal combustion engines shall be 0.003, 0.005, 0.010, 0.015, and 0.030. The standard oversizes for tractor, truck, and industrial internal combustion engines shall be 0.010, 0.020, 0.030, and 0.040. Large oversizes, when necessary, shall be held to 0.010 in.

"Piston rings shall be held to the same oversizes, omitting the 0.003-in. oversize, as are specified for pistons."

■

In passenger-car nomenclature: There is a definite trend toward the use of the "phaeton" in place of "touring car."

■

In a steam-cooling system, according to Alexander Herreschoff, manager, Rushmore Laboratory of Plainfield, freezing can be prevented by the addition of alcohol, the percentage of which remains constant all winter because it is condensed and returned in the same manner as is the water. Other advantages of steam cooling include use of a heater for warming the car body and freedom from noise, odor and fire hazards.

■

A 4-stroke-cycle marine-type diesel engine, according to Philip L. Scott, Diesel Tractor Corp. of LaPorte, Ind., "is about 25% more expensive than the steam engine. The 2-stroke, double-acting diesel costs about $60 per hp, which is very close to the cost of steam power. With the 2-stroke cycle, he said, I think we can cut the cost below that of the present 4-stroke cycle diesel-engine, but that is simply a guess."

Statistical Quality Control as Cost Saver

Based on paper by

E. J. YOUNG, JR.

Ford Motor Co.

STATISTICAL quality control was first applied to machining operations by the Ford Motor Co., shortly after World War II. Since then it has been used throughout manufacturing and assembling operations. Percentage defective, or "P" charts, are now used in nine locations of the plant to tell the exact status of quality at the end of each hour of the day.

Indicative of the savings made possible by this control, $20,000 was saved on the engine push rod operation alone within a year's time.

In August, 1949, approximately 60% of the units coming off the final assembly line were o.k., that is, 60 out of every 100 cars delivered to dealers were without need of repair or correction of defects. Now, 80% of units are o.k., despite a substantial increase in output.

In September, 1949, dealers were reporting better than an average of five defects per car, which cost the company an average of $2.60 to repair. A similar report as of May, 1950, gave the average number of defects as 2.6 and the cost to repair $0.82.

The company is now attempting to get its vendors to adopt statistical quality control. In most purchase agreements of the past it has been assumed that vendor shipments would be completely acceptable, but a more common sense approach to getting high quality in purchased parts is, first—to find out what quality level the vendor is able to achieve, and second—work with him to improve that level through quality control.

The paper also discusses the funda-

Continued on Page 86

TECHNICAL COMMITTEE *Progress*

SAE
Technical Board
W. H. Graves, Chairman

B. B. Bachman
Harry Bernard
G. W. Brady
A. T. Colwell
G. A. Delaney
C. T. Doman
Charles Froesch
C. E. Frudden
L. A. Gilmer
A. G. Herreshoff
R. P. Kroon
R. P. Lansing
Arthur Nutt
R. J. S. Pigott
W. D. Reese
H. L. Rittenhouse
R. R. Teetor
R. L. Weider
D. K. Wilson
H. T. Youngren

Technishorts . . . .

SCREW THREADS: One company found that Unified screw thread tolerance classes 2A and 2B cost less to make and use than the old classes of fit, it was recently reported at an SAE Screw Threads Committee meeting. Taking Class 2A cost factor as 100, this company estimated the cost factor of Class 2 external coarse threads to be 100; Class 2 external fine threads, 125; Class 3 external coarse threads, 150; Class 3 external fine threads, 200. With Class 2B cost factor as 100, the estimate showed the Class 2 internal coarse thread cost factor is 125; Class 2 internal fine thread, 150; Class 3 internal coarse threads, 200; and Class 3 internal fine threads, 250.

SPARK-PLUG CLEANING: A chemical cleaning process for aircraft spark plugs, developed by Auburn Spark Plug Co., is said to do an unusually good job. According to A. J. Battey, of Auburn, tests show complete enough deposit removal after cleaning to give infinite resistance. At a recent meeting of the SAE Ignition Research Committee, he said the process consists of: (1) degreasing plugs; (2) coating various plug parts with a stop-off lacquer, which curbs corrosion of platinum parts by the cleaning solution; (3) capping plug top to keep cleaning solution outside plug well; (4) immersing in solution for 20 to 30 min; (5) rinsing in hot water; and (6) preserving plug. The stop-off lacquer need not be removed since it seems to have no affect on plug operation. Battey offered to clean aircraft spark plugs for anyone interested in the process or desiring to make checks on possible rating changes caused by the cleaning process.

AUTOMOTIVE SEATING: Scope and program of the newly formed Automotive Seating Subcommittee, of the SAE Body Engineering Committee, have been crystallized. Objectives of the group are: (1) to standardize names of parts such as seat adjusters, seat frames, padding, and fastenings; (2) to develop methods of testing these parts; and (3) to develop recommended practices covering material and construction specifications, installation, and function. E. C. Pickard, Ford Motor Co., heads the new group.

ASA COMMITTEES: Two ASA committees are changing their scope and one of these also is changing its name. ASA Sectional Committee B27 on Standardization of Plain and Lock Washers will be renamed Standardization of Washers and Machine Rings, B27. Its scope is being expanded to include retainer rings. ASA Sectional Committee B29, Roller Chains, Sprockets, and Cutters will now encompass detachable link type chains. The co-sponsors of both these groups, ASME and SAE, have okayed the changes. They await final approval by ASA.

INTERNATIONAL STANDARDS: P. J. Kent, Chrysler Corp., will attend a meeting of Technical Committee 22, of the International Organization for Standardization, in Paris, Oct. 9 to 14, as a representative of both SAE and the Automobile Manufacturers Association. Among the items to be discussed are specifications for passenger car lighting, spark plugs, and brakes.

SAE to Standardize Plane Electric Motors

SAE standards and specifications planned for a complete line of fractional horsepower electric motors for aircraft promise inventory savings for users and manufacturing economies for producers. The SAE Aircraft Electrical Equipment Committee aims to standardize ratings, flange dimensions, speeds, and torque of these motors.

Committee Chairman T. A. Weiss, Republic Aviation Corp., reports that favorable comment on the program, from engineers surveyed, led to its adoption.

For example, an airline engineer sees such standards leading to lowered spare parts requirements and decreased overhaul cost due to fewer types. Another engineer feels standardized electric motors would allow greater interchangeability of different manufacturers' motors on the same equipment. He predicted elimination of current difficulties in getting quotations and preparing quotations and designs. Testing cost also should be reduced.

An electrical manufacturing man

T. A. Weiss, Chairman, SAE Aircraft Electrical Equipment Committee

agreed that fewer motor models make it easier for both manufacturing and service. Lowered cost should stem from increased volume, more production from the same facilities, and better parts availability.

Other gains cited were simplification of stocking and eventual reduction in inventory requirements; easing of the engineering design load; and aid to military procurement, particularly in this period of mushrooming military production.

The Committee, says Chairman Weiss, recognizes too the possible shortcomings of such motor standards, but doesn't feel that they are serious. In certain cases, selection of a standard motor may be a compromise and may penalize weight. Perhaps a designer may find no standard motor satisfying both space and torque requirements. In such special instance, he will have to resort to a nonstandard design. But that's no disadvantage brought on by the standard, since at present practically every motor purchased is a special.

The Committee feels assured that its new standardization program should not lead to motor design stagnation. It plans to keep abreast of latest technical developments and to incorporate them into its standards if practice so dictates.

SAE was requested to tackle the standardization of aircraft electric motors by the Aircraft Industries Association.

Revised AMS

The following revised Specifications are being circulated to industry for comment and criticism by the SAE Aeronautical Materials Specifications Division.

They are:

- AMS 2231A, Tolerances, Carbon Steel Bars
- AMS 2251A, Tolerances, Alloy Steel Bars
- AMS 2263A, Tolerances, Nickel and Nickel-Base Alloy Tubing
- AMS 5010C, Steel, Screw Stock (SAE 1112)
- AMS 5022E, Steel—Free Cutting (0.14-0.20C) (SAE 1117)
- AMS 5030A, Steel Wire, Welding, Low Carbon
- AMS 5036B, Steel Sheet and Strip, Aluminum Coated, Low Carbon
- AMS 5050D, Steel Tubing, Seamless, Low Carbon (SAE 1010), Annealed
- AMS 5061A, Steel—Low Carbon
- AMS 5062A, Steel, Low Carbon
- AMS 5075A, Steel Tubing, .22-.28 Carbon (SAE 1025)
- AMS 5080B, Steel, 0.32-0.38C (SAE 1035)
- AMS 5110A, Music Wire, Commercial
- AMS 5132C, Steel, High Carbon

More Accurate Formula Found For Cetane Number

A more acurate method of calculating ASTM cetane number of diesel fuels has been developed by a group of the CRC Diesel Fuels Division. Described in the newly-released CRC report,[1] "Methods for Estimating Cetane Number," the new method is designated the CFR Calculated Cetane Index.

Within the 40 to 60 cetane number range of commercial diesel fuels, expected accuracy of the index is close to ± 2 cetane numbers for 75% of the fuels to which it is applied. For fuels above 60 cetane, about 50% can be estimated within ± 4 cetane numbers.

The index is particularly applicable to straight-run diesel fuels, catalytically cracked fuels, and blends of the two.

The new method actually is a duel refinement of the calculated cetane number, one of five other ways used for estimating diesel fuel ignition quality. The two modifications are statistical—one for linear regression and the other for curvilinear regression.

Expression for the new index is:

CFR Calculated Cetane Index =

97.833 (log Mid-Boiling Point, °F)[2]

[1] Paper based on this report has been presented to the American Petroleum Institute.

Effects of Sulfur
As Shown in Lab Tests . . . .

HIGH-SULFUR gasolines cause more engine wear than those with low sulfur, a CRC dynamometer test program confirmed. It also showed that high engine operating temperatures retard wear rates. These and other results of this test program are spelled out in a report by the CFR Motor Fuels Division of the CRC, "Effect of Fuel Sulfur Content on Engine Wear and Corrosion in Laboratory Dynamometer Tests."

The report points out that fuels with 0.24 to 0.29% sulfur content gave more than twice as much piston ring and cylinder wall wear as fuels containing 0.05% sulfur. This ratio was found at both low and high engine operating temperatures. But with either type fuel, increasing operating temperature greatly reduced wear.

These tests are said to corroborate the thinking that corrosive cylinder wear is related to jacket temperature. Low jacket temperature brings high wear, high jacket temperature lowers wear. For example, in one engine the decrease in jacket temperature from 160 to 80 F about tripled wear. In another engine, a temperature reduction of from 212 to 95 F increased wear more than eight times.

In fact low engine temperatures with low sulfur content fuels gave more wear than high engine temperatures with high sulfur fuels.

The report also says that at low operating temperatures, some oils meeting U. S. Army Spcification 2-104B somewhat reduce ring wear, engine parts corrosion, and engine deposit formation. However, tests showed no effect of sulfur on engine deposits at either high or low temperature.

Various pure sulfur compounds were added to the fuels during the tests. But engines cannot differentiate between the sulfur compounds, this study showed. Effect on wear is about the same regardless of the sulfur compound in the gasoline. One laboratory studying various sulfur compounds obtained some interesting data. They showed a linear relationship between gasoline sulfur content and engine wear, for sulfur contents of practically zero to 0.50%.

Apparently fuel factors other than sulfur content affect engine wear and corrosion. Available data indicate that straight run fuels give more wear than cracked fuels. They also showed that adding tetraethyl lead to straight run fuels increases wear; but such additions to cracked fuels seemed to have no effect on wear.

These dynamometer test results served as a guide in setting up a field test program on effects of sulfur in gasoline. For these field tests (reported at right), the report recommends a fuel meeting these specifications: 0.25 to 0.35% sulfur; 70 to 80 octane number, ASTM; a tetraethyl lead content within 10% of the reference fuel against which the test fuel will be compared; volatility suitable for the operation involved; and a composition consisting of 40 to 60% cracked gasoline and the balance straight run.

The report, CRC-249, has 70 $8\frac{1}{2} \times 11$ pages, including 12 charts and photographs. It is available from the SAE Special Publications Department. Price: $2.00 to members, $4.00 to nonmembers.

+ 2.21 (API Gravity) (log Mid-Boiling Point, °F) + 0.0125 (API Gravity)^2 − 423.5 (log Mid-Boiling Point, °F) − 4.78 (API Gravity) + 419.63

The report contains an alignment chart or nomograph for this equation. Knowing API Gravity and Mid-Boiling Point, it is possible to determine quickly the CFR Calculated Cetane Index from this chart without going through the mathematical manipulations in the formula.

There are three limitations to the use of the CFR Calculated Cetane Index for estimating ASTM cetane number. They are:

1. It is not applicable to fuels containing additives for increasing cetane number.
2. It is not applicable to pure hydrocarbons, synthetic fuels, fuels produced by alkylates, and coal tar products.
3. It may give considerable inaccuracies if applied to crude oil, residual fuels, or fuels of below 400 F end point.

The CFR Calculated Cetane Index is a tool for estimating ASTM cetane number, says the report. But it cautions that in view of its limitations, the index cannot be substituted for the ASTM cetane number determined in a test engine.

The indexes evaluated in this program covered 638 fuels; 980,000 separate calculations were made in the analysis; and it took 990 man-hours to complete the job.

The data in the report were analyzed and prepared by a panel consisting of: H. D. Young, Sinclair Refining Co., leader; Royce Childs, Waukesha Motor Co.; H. M. Gadebusch, Detroit Diesel Engine Division, GMC; R. P. Gilmartin, Gulf Research & Development Co.; F. L. Nelson, Socony Vacuum Laboratories; and W. K. Simpson, Electro-Motive Division, GMC.

The report, CRC-251 has 33 8½ × 11 pages, including 14 tables and 12 charts. It is available from the SAE Special Publications Department. Price: $1.00 to members, $2.00 to non-members.

in Gasoline

. . . . As Shown in Field Tests

INCREASING sulfur content has less of an effect on engine wear in commercial vehicle engines in heavy-duty operation than in smaller truck and passenger car engines in light-duty operation. These and other facts emerged from field service tests of 62 vehicles, which included 10 engine and vehicle types) in seven fleets, covering more than one and one-half million miles. Two motor gasoline sulfur levels —0.056 to 0.140% and 0.25 to 0.30%, were compared for their effect on engine wear and cleanliness.

These test results are analyzed and interpreted in a recently released CRC report. "Sulfur in Motor Gasoline," prepared by a group under the CFR Motor Fuels Division.

A significant finding discussed in the report is that increasing fuel sulfur, over the range shown in Table 1, produced essentially no change in engine wear in three fleets. These vehicles included four engine types operating in the following classes of service:

a. Fleet 1—heavy-duty urban bus service.

b. Fleet 2—heavy-duty over-the-road truck.

c. Fleet 7—medium-duty bus and light truck service.

Table 1 shows the wear data.

In another fleet, increasing fuel sulfur content from 0.056 to 0.269% markedly increased wear. The one type engine involved here was engaged in medium-duty intermittent delivery service, typical of city door-to-door operation. Using heavy-duty crankcase oil in this fleet reduced cylinder wear 50% in vehicles operated on low sulfur fuel; but it did not appreciably reduce cylinder wear in the high sulfur vehicles.

Consistent increases in wear were brought about by raising fuel sulfur content in vehicles of four fleets operating in light-duty intermittent service. Results obtained here are shown in Table 2.

Effect of engine design on difference in wear due to sulfur cannot be separated out, notes the report, because of the influence of type of service. But in one fleet, using two vehicle types, percentages increase in wear with high sulfur were not significantly different. However, absolute wear rate was twice as great in one vehicle type as in the other, with both low and high sulfur fuels. Biggest difference in bore wear was 0.0016 in. over a test mileage of 2900 to 3400.

The report also says that high sulfur fuels did not seem to increase bearing or journal wear, with the exception of the vehicles in one fleet and one vehicle in a second fleet. Nor were any adverse effects on valve operation and life noted with high sulfur fuels. Three low-sulfur vehicles in one heavy-duty fleet were overhauled during the test period because of sticking valves. And high sulfur fuel did not reduce muffler life or corrode the exhaust system.

Interestingly enough, combustion chamber deposit weights were roughly the same for low and high sulfur vehicles. In all but one of the seven fleets, high sulfur had little or no effect on engine varnish and slude deposits. One fleet, in door-to-door delivery service showed differences of 7 to 10 points (out of a possible 100 in overall deposit rating) in favor of the low sulfur fuel.

Heavy-duty oil was compared with

Continued on Page 82

Table 1—Summary of Engine Cylinder and Ring Wear in Medium and Heavy-Duty Service

Fleet	Vehicle Type	Fuel S Content Wt. % Low S	Fuel S Content Wt. % High S	% Increase in Wear, High S over Low S (Corrected for Mileage) Cyl	Ring
1	A	0.114	0.256	−14	14
2	B	0.091	0.281	−18	−2
7	F	0.132	0.264	10	6
	HA	0.132	0.264	−43	7
	J	0.132	0.264	0	29

Table 2—Summary of Engine Cylinder and Ring Wear in Light-Duty Intermittent Service

Fleet	Vehicle Type	Fuel S Content Wt. % Low S	Fuel S Content Wt. % High S	% Increase in Wear, High S over Low S (Corrected for Mileage) Cyl	Ring
3	C	0.063	0.267	110	68
4	D	0.066	0.267	80	111
6	D	0.140	0.290	75	56
6	G	0.140	0.290	88	31
7	H	0.132	0.264	65	44
7	I	0.132	0.264	123	92

BRIG.-GEN. CLYDE H. MITCHELL, commanding general of the 52nd Fighter Wing, New York Air National Guard, has been recalled to active duty with the United States Air Force. He is to head the deferment board at Headquarters, Continental Air Command which handles appeals of Air Force reservists ordered to active duty. In civilian life, General Mitchell was district sales manager for the Gates Rubber Co., New York.

COL. J. G. VINCENT, SAE president in 1920, will retire December 31 from his position as Packard Motor Car Co.'s executive vice-president. He will continue as a director, and will hold the new title of engineering consultant after his retirement. Colonel Vincent has been with Packard for over 38 years, starting in 1912 as chief engineer. After his appointment as vice-president of engineering in 1915, he directed all the design, research, and other developmental work on the company's automotive, marine engine, and aircraft engine projects. He designed the first 12-cylinder engine, the Packard Twin-Six, in 1915, supervised creation of the first successful diesel engine for aircraft, and co-designed the Liberty Engine in World War I. He developed the Packard marine engines used in the PT boats of World War II, and contributed much to the improved design of the Rolls-Royce aircraft engine which Packard built for five types of fighting planes. He became a Packard director in 1945, and executive vice-president in 1949.

FRANK N. PIASECKI, board chairman of the Piasecki Helicopter Corp., Morton, Pa., has been elected to the chairmanship of the Helicopter Council, Aircraft Industries Association of America, at Washington, D. C.

JOHN B. TAYLOR, JR., has been appointed assistant director of research at the Ethyl Corp. Research Laboratories, Ferndale, Mich. to direct research on automotive and aviation products. Taylor joined Ethyl 19 years ago as a field engineer in Baltimore, Md., and three years later was transferred to the company's technical service activities in Detroit. During the war, he served as coordinator and assistant director of engineering research, and for the past three years has been executive engineer for the research laboratories.

CADILLAC APPOINTMENTS

EDWARD N. COLE, works manager of the Cadillac Motor Division, GMC, has been selected to serve as plant manager for the new Cadillac Motor Car Division—Cleveland Tank Plant, Cleveland, Ohio. Cole is SAE Vice-President representing Passenger Car Activity.

H. F. BARR has been named chief engineer at the new plant. Prior to this, he was assistant chief engineer with Cadillac in Detroit.

About

C. HUNTER LINDSAY is now associated with McCarty's Oakland, Calif. He is responsible for the production of McCarty's products, which include automobile seat covers, tires, and so forth. Formerly, Lindsay was vice-president in charge of engineering with the Lindsay Corp., Melrose Park, Ill.

GEORGE J. SCRANTON, formerly manager of the inspection department for the Ford Motor Co., Dearborn, Mich., is now manager of the standards and methods department with Ford. His new position entails functional supervision of inspection and quality standards inspection methods, gage and layout process and liaison with engineering.

C. E. WILLIS has been appointed assistant sales manager of the Electro-Mechanical Division, Lear, Inc., Grand Rapids, Mich., and **A. N. LAWRENCE** appointed manager of the Teterboro, N. J. eastern office of the same company. Willis has been manager of Lear's eastern office since 1947. Upon assuming his new duties of directing Lear west coast sales and field engineering activities, he will make his headquarters at the company's Los Angeles plant. Lawrence, formerly aviation application engineering department manager for Jack & Heintz, and a graduate of both Princeton University and M. I. T., brings to Lear a successful background in aircraft equipment sales engineering.

The National Committee for the USA for the Third World Petroleum Congress Meeting in The Hague on May 28-June 6, 1951 has completed its organization including **K. G. MACKENZIE,** The Texas Co., New York as secretary, and **EARL BARTHOLOMEW,** Ethyl Corp., Detroit, as a member to the executive committee.

Members

VAN M. DARSEY, president of the Parker Rust Proof Co., Detroit, has been elevated to vice-chairman of the board. **F. J. DeWITT, JR.**, general sales manager, has been appointed vice-president in charge of sales. Darsey, who joined the company in 1927, was technical director for several years before becoming president. As vice-chairman he will devote much of his time to product development and line expansion projects.

THEODORE H. ELIADES, previously a junior mechanical engineer with U. S. Gypsum Co., Clark, N. J., is now an accountant with Compressed Air Products, Maplewood, N. J.

L. B. EBBS has retired from the GMC Truck & Coach Division, Boston Zone, and is acting as jobber for the New England area of the Industrial Chemical Co., Los Angeles, Calif.

WYMER G. VOGEL, formerly a junior engineer with the Nebraska Department of Roads and Irrigation, Lincoln, Nebr., is now employed by the Chance Vought Aircraft Division, United Aircraft Corp., Dallas, Texas, in the capacity of loftsman in the engineering department.

JAMES R. GLYNN is now employed in the powerplant unit of the Boeing Airplane Co., Seattle, Wash., as a junior engineer "A". Prior to this, he was a test engineer with Pratt & Whitney Aircraft, Hartford, Conn.

ROBERT V. ROSENWALD is now connected with the Minneapolis-Honeywell Regulator Co., as a design engineer in the aeronautical engineering department. Previously, he was a mechanical engineer at the U. S. Naval Engineering Experiment Station, Annapolis, Md.

CYRUS R. OSBORN, vice-president of General Motors and general manager of the Electro-Motive Division, is now group executive in charge of the general engine divisions. He is also a member of the administration committee. First employed by General Motors as an apprentice in the Dayton Engineering Laboratories Co. in 1921, Osborn then entered the overseas field in 1923. After serving in various executive capacities abroad and in the United States, he became assistant in the general engines group in 1941. He served briefly as general manager of the Allison Division in 1943, and in the same year was elected a vice-president of General Motors and appointed general manager of Electro-Motive Division.

J. H. COOLIDGE, vice-president and treasurer of Thompson Products, Inc., Cleveland, Ohio, has been elected president of the Ramsey Corp., which was recently purchased by Thompson Products. **MELVIN W. MARIEN**, formerly chief engineer with the Ramsey Corp., is now vice-president of engineering with the new subsidiary.

C. R. MAXON of the New Jersey Zinc Co. was presented with the First Annual Doehler Award of the American Die Casting Institute, New York. This award, made each year "for the outstanding achievement contributing to the advancement of the die casting industry," was a popular recognition of Maxon's accomplishments while working with federal agencies in Washington and the Canal Zone earlier this year.

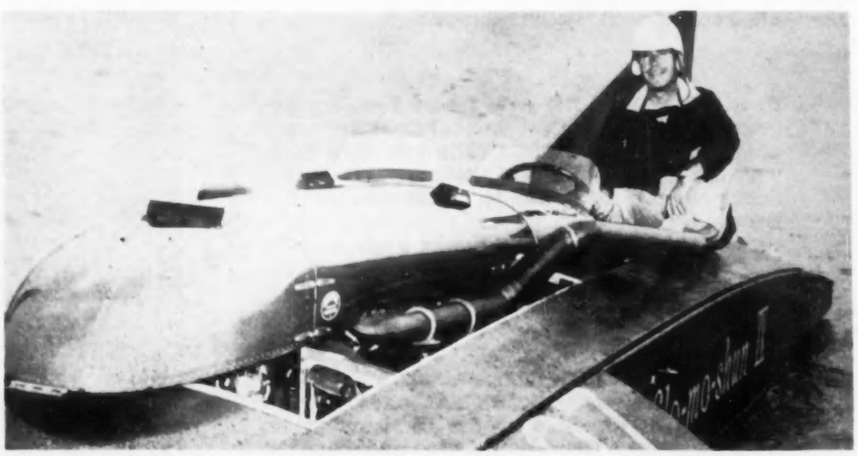
ELMER W. KRUEGER, operations manager at the Cleveland Pneumatic Tool Co., was elected to the board of directors of the company at its recent meeting. Krueger was born and educated in Cleveland and came with the Pneumatic Tool Co. in 1922, and was appointed operations manager in June, 1949.

ELMER R. BARTOSEK, formerly an engineer in the engines and lubricants section of the Armour Research Foundation, Chicago, Ill., is now connected with Chance Vought Aircraft Corp., Dallas, Texas, in the capacity of engineer, powerplant analysis. His new position entails the complete analysis of powerplant from the performance aspect under all flying conditions. He was student chairman of the Illinois Institute of Technology Branch, 1948-49.

HENRI PERROT has been chosen president, Societe des Ingenieurs de L'Automobile for 1950-51. A veteran of the automobile industry, Perrot received his engineer's degree in arts and sciences in 1902, and has made frequent and important design contributions since, particularly in the brake field. He took his first patents on braking devices in 1910, and his early pioneering has contributed much to what has been accomplished in the braking industry. In 1923 he joined Vincent Bendix to form the Perrot Brake Corp., which became Bendix Brake Corp. and eventually Bendix Aviation. Perrot joined SIA in 1927, and has been treasurer and vice-president. He has been an SAE member since 1924.

ROBERT L. STANLEY joined the staff of the Diesel Engine Manufacturers Association on October 1 as educational director. Professor Stanley came to this position from The Pennsylvania State College where he has been working in diesel engine research in the Engineering Experiment Station.

DAVID ROY SHOULTS will direct testing at the Air Force's Arnold Engineering Development Center, near Tullahoma, Tenn., for Aro, Inc. Aro recently was awarded a contract for operating this government research and development facility. Shoults formerly was vice-president of engineering for the Glenn L. Martin Co., and also has been associated with Bell Aircraft Corp. and General Electric Co.



L. J. (LOU) FAGEOL, president of Twin Coach Co., with "Slo-Mo-Shun IV," the two-ton, 29-ft hydroplane that enabled him to win the International Harmsworth Trophy with a new record average speed of 100.68 mph.

VERNE H. SCHNEE, director of t University of Oaklahoma Research I stitute, has been appointed vice-pres dent in charge of development at t university. The new post was ma to help meet educational proble accompanying the Korean cris Schnee also holds the title of profess of metallurgy at the university.

WARREN W. JONES, SR., former chief engineer with the Liggett Sprin & Axle Co., Monongahela, Pa., is no employed by the Sun Electric Corp Chicago, Ill. as zone representative His new position entails the sellin and servicing of equipment and training operators.

JEROME HUNSAKER, head of aeronautical engineering department at M. I. T., and **R. DIXON SPEAS**, U. S Representative, A. V. Roe, Canada, Ltd., were active participants in the Massachusetts Institute of Technology conference on Ground Facilities for Air Transportation, held at the Institute in Cambridge, Mass., September 12 to 14.

LAWLER B. REEVES, former assistant director of manufacturer's sales with the U. S. Rubber Co., Tire Division, Detroit, Mich., has been given the new assignment of manager of government sales, with offices at the company's Detroit plant. Reeves became employed by the company in Oklahoma City in 1941. He is a graduate of the University of Texas and saw action as a combat pilot in the last war, attached to the 9th Air Force with the rank of Colonel, USAF.

C. O. SLEMMONS, formerly chassis engineer with the Studebaker Corp., South Bend, Ind., is now employed by The General Tire & Rubber Co., Akron, Ohio.

WILLIAM E. McCARTHY is now working for Ford Motor Co., Dearborn, Mich., as a technician in the Research Laboratory. Prior to this, he was an engineer at the General Motors Proving Ground, Milford, Mich.

WILLIAM P. KALB, previously a sales engineer with the Skinner Purifiers Division, Bendix Aviation Corp., Detroit, Mich., is now employed by Lorr Laboratories, Inc., Patterson, N. J. as a sales executive. His new position entails the coordination and liaison between the sales department and the manufacturing department.

ROBERT T. STROUSE, formerly a district bus manager with Mack International Motor Truck Co., St. Paul, Minn., is now president of Charlie's Motors, Inc., Montevideo, Minn.

MILTON D. ADLER is now a mechanical engineer with the Viking Air Conditioning Co., Cleveland, Ohio. Prior to this, he was a project engineer with the Jacobs Aircraft Engine Co., Pottstown, Pa.

Students Enter Industry

ELBERT L. WILCOX (Yale University '50) to Aluminum Co. of America, Edgewater, N. J.

HUGH A. WILLIAMS, JR. (North Carolina State College '50) to Lima-Hamilton Corp., Hamilton, Ohio.

EDWARD S. BAKER (Indiana Institute of Technology '50) to South Carolina Highway Department, Chesterfield, S. C.

PAUL EMELIAN (Detroit Institute of Technology '50) to H & A Tool & Die Co., Detroit, Mich.

JAMES B. MCKEON (Michigan State College '50) to U. S. Radiator Corp., Detroit, Mich.

JAMES E. RIDGWAY (Indiana Technical College '50) to Ocean City Mfg. Co., Philadelphia.

DALE T. KURLINSKI (Ohio State University '50) to The Hoover Co., North Canton, Ohio.

ROBERT E. LEE (Tri-State College '50) to Union Steel Products Co., Albion, Mich.

CARROLL FREEMAN (The Aeronautical University '50) to ServAero Corp., Miami.

EDMUND R. HINMAN (Northrop Aeronautical Institute '50) to United Aircraft Corp., East Hartford, Conn.

JOHN E. MAZUROWSKI, JR. (Rensselaer Polytechnic Institute '50) to American Machine & Foundry Co., Buffalo, N. Y.

CHARLES W. MERVINE (Chrysler Institute of Engineering '50) to Chrysler Corp., Highland Park, Mich.

HARVEY E. MOOSE (Purdue University '50) to Boeing Airplane Co., Seattle, Wash.

ROBERT W. SLIGER (University of Idaho '50) to Allis-Chalmers Mfg. Co., Milwaukee.

ALBERT RICHARD EDSON (Rensselaer Polytechnic Institute '50) to Owens-Corning Fiberglass Corp., Newark, Ohio.

FRANCIS V. MUSHIAL (Purdue University '50) to White Motor Co., Cleveland.

ELISEO R. ROSA (Rensselaer Polytechnic Institute '50) to Singer Mfg. Corp., Elizabethport, N. J.

JOHN C. FISCHER, JR. (University of Wisconsin '50) to Consolidated Vultee Aircraft Corp., Fort Worth, Texas.

WILLIAM T. GREENE (University of Virginia '50) to General Motors Corp., Detroit, Mich.

GENE STANLEY LUFF (University of Oklahoma '50) to Coston & Frankfurt, Oklahoma City.

FREDERICK H. IMMEN (Cornell University '50) to Boeing Airplane Co., Seattle, Wash.

ALVIN R. GARLING (Oklahoma A & M College '50) to Socony Vacuum Oil Co., Inc., Augusta, Kans.

RICHARD F. DAVIS (Michigan State College '50) to White Products Co., Middleville, Mich.

VICTOR G. BAKER, JR. (University of Colorado '50) to Refinery Engineering Co., Tulsa.

WILLIAM L. BEARD, JR. (Yale University '50) to R. R. Donnelley & Sons Co., Chicago.

RAY C. SLAY, JR. (University of Oklahoma '50) to Consolidated Vultee Aircraft Corp., Fort Worth, Texas.

KENNETH G. PRIBBLE (University of Illinois '50) to Central Foundry Division, Tilton, Ill.

KENNETH D. ROBERTS (Fenn College '49) to Oildex Sales Co., Colorado Springs.

GLENN B. SHEW (Northrop Aeronautical Institute '50) to Douglas Aircraft, Long Beach, Calif.

JOSEPH F. PETROSIUS (Illinois Institute of Technology '50) to Associated Valve & Engineering Co., Chicago.

LELAND C. TUPPER (Washington State College '50) to General Electric Co., Schenectady, N. Y.

CHARLES R. HARPER (General Motors Institute '50) to Central Cadillac Co., Cleveland, Ohio.

ALLIE O. ISOM (University of Oklahoma '50) to Fruehauf Trailer Co., Oklahoma City.

KEITH N. WATTS (Stanford University '50) to Boeing Airplane Co., Seattle, Wash.

HAROLD T. KERNS (University of Illinois '50) to The Buda Co., Harvey, Ill.

STANLEY F. KINNE (Oregon State College '50) to Peerless Trailer & Truck Service, Inc., Portland, Oreg.

FRANK I. LAUSH (University of Washington '50) to Puget Sound Sheet Metal Works, Seattle, Wash.

JAN EZRA PETERSEN (Rensselaer Polytechnic Institute '50) to Ezra Petersen Autostores, Fredericia, Denmark.

LAWRENCE C. FITCH (Marquette University '50) to AC Spark Plug Division, GMC, Milwaukee, Wis.

ROBERT, J. PETERSON (University of Wisconsin '50) to Kohler Co., Kohler, Wis.

ALBERT W. BITZER (Carnegie Institute of Technology '50) to Heintz Mfg. Co., Philadelphia.

CHARLES JOHN KLAMBT (Yale University '50) to Torrington Co., Torrington, Conn.

WILLIAM ALFRED PRICE (Purdue University '50) to Price Garage, South Bend.

EDWARD J. KLICH (University of Illinois '50) to Fruehauf Trailer Co., Chicago.

RAY A. VAN DE WALKER (Northrop Aeronautical Institute '50) to North American Aviation, Inc., Inglewood, Calif.

KENNETH A. LEONARD (University of Colorado '50) to Broadmoor Hotel, Colorado Springs, as assistant engineer.

ROBERT C. JUVINALL, former supervisor of the engineering staff of the Chrysler Corp. research design department, has been named associate professor of mechanical engineering at Illinois Institute of Technology. From 1945 to 1948, Juvinall lectured on combustion at Chrysler Institute, and taught machine design in 1947 and 1948 in the Wayne University evening school. While working for his master's degree at Illinois, he has been an assistant professor in machine design.

RICHARD L. BERRY, formerly with the Detroit office of the American Brake Shoe Co., has been transferred to the Chicago office. He is a sales engineer.

L. T. BARNES left General Motors of Canada on September 1 to assume the position of sales manager, Waterous Ltd., Edmonton, Alberta, Canada.

WILLIAM E. HERBY, formerly design engineer with The Gray Marine Motor Co., Detroit, is now employed as a special development engineer with The White Motor Co., Cleveland, Ohio.

DR. GUSTAV EGLOFF, director of research, Universal Oil Products Co., Chicago, Ill., has been elected a Fellow of the Royal Society of Arts of Great Britain.

CHARLES-ROBERT PEARSON, formerly an assistant professor of aeronautical engineering at the University of Florida, is now employed by the Boeing Airplane Co., Seattle, Wash., in the capacity of field service engineer.

RUSSELL T. HOWE, who, prior to this, was factory manager with R. G. Le Tourneau, Inc., Vicksburg, Miss., is now connected with A. M. Kinney Engineers Associates of Cincinnati, in the capacity of vice-president.

L. A. STEWART, formerly chief engineer with the American Coach & Body Co., Cleveland, Ohio, is now employed by the Motor Products Corp., Detroit.

D. FRANKLIN BOYD, who, prior to this, was quality control analyst with the Kaiser-Frazer Corp., Detroit, Mich., is now manufacturing analyst for the Ford Motor Co., Dearborn, Mich.

PAUL C. PERRY is now district manager for the Union Oil Co. of California in the Canal Zone. Prior to this, he was technical advisor, foreign sales for that same company in Los Angeles, Calif.

ALVIN P. WILLIAMS, JR., is now a technical writer in the Service Engineering Division of the Beech Aircraft Corp., Wichita, Kans.

Continued on Page 96

OBITUARIES

ROBERT T. HAZELL

Robert T. Hazell, one of the best known figures in the Canadian automotive truck and trailer field, and vice-president and general manager of the Fruehauf Trailer Co. of Canada, Ltd., died at his home on January 14. Born in Hamilton, Ontario, he attended Lake Lodge School, and entered the truck business after leaving high school. He joined Fruehauf as sales manager in March 1948, and in June, 1949 was appointed vice-president and general manager. Mr. Hazell took an active part in the Automotive Transport Association and was a former director of that organization. He was chairman of the 1949 National Truck Roadeo Committee, and was also a past president of the Kiwanis Club of Winnipeg and an honorary member of the Kiwanis Club of Toronto.

ROBERT COLLINS

Robert Collins, a research engineer in the General Motors Research Laboratories, was killed in a midget auto race at Lake Simcoe, Ontario, when the rear wheel of his car collapsed and the vehicle overturned on him. Collins raced midget cars as a hobby.

SIGMUND LISBON

Sigmund Lisbon, chief design engineer with the International-Plainfield Motor Co., Plainfield, N. J., passed away on April 25. He was 47 years old.

He was born in Providence, R. I., and attended the Technical High School and the Rhode Island School of Design, both located in that city. His association with the Mack Company started on September 14, 1925, and he was still employed there at the time of his death. Lisbon enjoyed sports, particularly golf, swimming, and bowling. He was especially fond of watching football and baseball, and was also developing television as a hobby.

GEORGE W. RUMFORD

George W. Rumford, a pioneer automotive industry executive, died August 2 at his home in Detroit. He was 68.

Rumford, whose industrial career spanned a half-century, was operating manager of the Chrysler Corp's DeSoto plant at the time of his retirement in 1947. After his retirement, he became active as a partner in the George Rumford Motor Sales Co., Detroit. Born in Forrest, Ontario, he moved to Port Huron at 16 and entered the employ of the Engine and Thresher Works. From then until his death, he held various positions in the automotive industry.

MARK B. HARRIS

Mark B. Harris, an SAE member for 26 years, died suddenly in September. He was 61 years old. An engineer with the Stewart-Warner Corp., Detroit, Mich., he had been associated with the old Oakland Motor Car Co. and the Franklin Motor Co.

WARREN W. DEWEY

Warren W. Dewey died Sunday, August 20, at Bethesda Naval Hospital of bulbar polio. Mr. Dewey was Washington sales representative for the Heil Co., a Milwaukee, Wis., heavy-machinery firm. He served with the Navy Supply Corps in the Pacific, reaching the rank of lieutenant (j.g.). Born in Kohler, Wis., Dewey was graduated from the University of Wisconsin. He was 29 years old.

FRANK L. MAURO

Frank L. Mauro passed away in July. He was a junior designer with Jack & Heintz Precision Industries, Inc., Bedford, Ohio. He had spent much of his time in the hospital because of a bone infection acquired when he was a child. For the past ten years he had been managing baseball teams consisting mainly of younger boys. He also managed a soft ball team for Possibilities Unlimited, a ball team consisting of men who are handicapped. He was 32 years old.

ALVIN F. KNOBLOCK

Alvin F. Knoblock, pioneer automobile executive and a resident of Detroit, Mich., for 65 years, passed away at his home on September 1, after a long illness. Mr. Knoblock, a graduate of Albion College, was a trustee of the college for many years. He was chairman of the board of the Bundy Tubing Co. and the Auto Glass Mfg. Co. He was pastor at Gratiot Methodist Church, and chairman of the board of trustees of the Metropolitan Methodist Church for 25 years. He was one of the first vice-presidents of General Motors Corp.

SAE AT LAWRENCE INSTITUTE

Plant trips, job clinics and cooperation with the Detroit Section are major elements in the active program which has helped Lawrence Institute of Technology's SAE Student Branch become the largest technical group on its campus—and one of the largest SAE Student Branches in the country.

Tours of factories and research laboratories have been focal points of student interest ever since the Branch's postwar program got into action. Back in 1947, Student Chairman Albert Nash set the pattern when he arranged the first of these information-producing visits to industry.

Since that time trips have been organized to plants of the United States Rubber Co., Detroit Tank Arsenal, Fisher Body, Gar Wood Industries, and many others. The program is arranged to permit an SAE Enrolled Student, during his four years at Lawrence Tech, to see for himself almost every phase of industrial work being conducted in the heart of the nation's greatest automotive manufacturing area.

Job-getting clinics, though less numerous than the plant visits, have also been important stimulators of value and interest to the Branch members. Started in 1949 under the guidance of Dr. Gail Brewington, the Branch Faculty Adviser, these groups have met to discuss and plan individual employment programs for graduates and to incorporate these plans into complete personal data sheets for their use. This clinical approach to the employment problems of the graduates has met with an enthusiastic response.

Participation in organizing the now-annual student-senior joint meetings held by the Detroit Section has benefited the Branch as a whole as well as those who have actively helped. First Branch Chairman Nash was responsible for getting this project under way. In the spring of 1947 the first of these meetings was held. Since proved popular, they are now a regular part of the Detroit Section program. At these affairs entertainment is provided by students during the dinner hour and some prominent speaker talks to the group afterwards. At the 1950 dinner, a skit was presented with a cast of students from the various schools in the Detroit area taking part. Following dinner, A. T. Colwell, vice-president, Thompson Products, Inc., spoke on "What Industry Expects of the Young Engineer."

These highlighted activities have been superimposed, of course, on a sound program of regular meetings, effective speakers, and special motion picture presentations at the Institute. Speakers participating in last year's program included engineers from such companies as Chevrolet, Cadillac, Thompson Products, Gar Wood Industries, and Chrysler Corp.

Faculty Adviser Brewington has been a steady guide to progress in Branch activity ever since interest in SAE first started at Lawrence way back in 1938. The night school men, who then dominated Lawrence enrollment, were interested but not too active until the beginning of World War II. Then, in 1946, in combination with a large day-school enrollment, they began to make things happen SAE-wise. The SAE Student Group grew to 150 members. It applied for a Student Branch charter. On April 11, 1947, the SAE Council said: "You're in."

And that's when Student Chairman Albert Nash started the multi-pointed program of activities which by 1950 brought the Lawrence Institute Stu-

Main Building of the Lawrence Campus

dent Branch to second place in membership among all SAE Student Branches.

1949-1950 Student Chairman Richard Nicholas carried on in the Nash tradition. "We're aiming to provide better technical fare for every Branch member all the time," he said last Spring. "We figure that the size of our membership will take care of itself, if all those who do belong get plenty of good engineering viands."

SAE MEMBERS WHO ATTENDED LAWRENCE INSTITUTE INCLUDE:

George E. Adams (**1937-42**), Leonard E. Adler (**1939-43**), Gilbert C. Amnotte (**1939-45**), Emmett W. Bond (**1940-43**), Ben F. Bregi (**1933-37**), William L. Casterline (**1942-44**), Carl W. Cowan (**1936-40**), William B. Crump (**1938-43**), Robert T. Curcuru (**1930-33**), Robert E. Davidson (**1948-49**), Edwin H. Donaldson (**1944-48**).

Edward Donley (**1939-43**), Leonard A. Dorr (**1929-30**), William F. S. Dowlding (**1933-39**), Norman J. Downey, Michael Durella (**1946-49**), Edward M. Eberhart, Jr., (**1941-48**), Herbert R. Fortgang (**1938-43**), Jack W. French (**1943-44**), Helma U. Fuhrmann (**1944-**), Chester A. Garbacz (**1939-42**).

Charles F. Gray (**1946-49**), Karl V. Holm (**1944-**), Edward H. Holtzkempes (**1937-38**), Emmett J. Horton (**1937-42**), Alexander Hossack (**1940-48**), William E. Jackson (**1939-41, 1944-45**), Louis J. Jelsch (**1935-44**), Lloyd E. Kamm (**1935-40**), Robert Keller (**1937-48**), Kenneth W. Kampman (**1932-33**).

W. G. Kinmont (**1933-35**), Michael B. Koswan (**1946-49**), John W. Krygier (**1939-45**), Roy G. LaGrant (**1937-42**), Theodore F. Lapinski (**1937-48**), Robert W. Lemon (**1945-49**), H. Douglas Lowrey (**1937-41**). Henry J. Malik, Jr. (**1946-50**), Robert R. Mandy (**1940-45**), Robert A. Meade (**1941-42**).

Walter B. Mills (**1941-48**), Robert M. Morkin (**1946-49**), Albert E. Nash (**1938-40, 1945-48**), Donald H. Nelson (**1938-44**), Robert C. Ofenstein (**1935-40**), Charles L. Payor (**1943-48**), Gordon A. Price (**1941-48**), Vernon E. Riddell (**1938-41**), Sydney Rogers (**1943-47**), Meyer L. Rothenberg (**1940-43**).

Arthur W. Rutkowski (**1941-48**), T. R. Schulz (**1938-40**), Joseph R. Sequin (**1937-44**), Karl Siegle (**1942-48**), Aram Sogoian (**1943-48**), John Taylor (**1932-35**), Kurt O. Tech (**1940-48**), Ralph E. Williams (**1939-41**), James T. Wilson (**1946-49**).

Why There Is a Field Editor

by W. F. SHERMAN

SAE Journal Field Editor, Detroit Section

SECTION activities in all of the areas where the SAE is active constitute a major part of SAE's annual over-all program. The technical papers presented before groups and sections are the primary methods of exchanging technical information, knowhow and ideas for which SAE is famous.

On no small scale the social activities, and the plain ordinary rubbing of elbows with fellow engineers, is important in the Society's program.

None of these could have a greater sounding board than the local meeting hall if it were not for the service of the numerous SAE Field Editors. Their main function is to prepare a concise report of each technical meeting for the SAE Journal. This gives the thousands of members in other sections a chance to find out what is going on in areas of main interest to them.

Along with this job, the Field Editor assists in making certain that copies of technical papers reach the New York office for consideration by the Editorial Committee, with a view to having them published in full if possible.

Meeting publicity photographs for use in the meeting reports are obtained by the Field Editor in cooperation with the Meeting Operations Chairman.

Being Field Editor is always a pleasant duty, for even though meeting attendance is a must for the Field Editor, it keeps him technically informed and in close touch with the friendly Section members. (From May issue of Detroit Supercharger.)

Twelve Past-Chairmen Attend Indiana Meeting

TWELVE past-chairmen of Indiana Section, one of whom held office in 1914, were present at the May meeting of Indiana Section. They are pictured below, with the years in which they served, together with past and present officials of the Indianapolis Speedway.

Left to right are: S. A. Silbermann; Wilbur Shaw; Macy O. Teetor (1938); Lon R. Smith (1914 and 1921); Chester S. Ricker; Roy W. Paton (1943); William S. Powell (1947); A. L. McColloum (1939); R. P. Atkinson (present acting chairman); Joseph Liston (1942); J. C. Miller, Jr. (speaker); Karl H. Effman (1946); R. M. Critchfield (1937); Lee Oldfield (1932); George L. Brinkworth (1941); William K. Creson (1935); Ray A. Schakel (1948); W. P. Wood (present treasurer).

SAE National

Diesel Engine Meeting

Hotel Knickerbocker, Chicago, Ill.

Nov. 2-3, 1950

Thursday, Nov. 2

9:00 a.m.

Welcome
—J. A. NELSON, General Chairman of Meeting

F. G. SHOEMAKER, Chairman

New Budd Diesel Railroad Car RDC-1 with Torque Converter Transmission

The Car
—BENJAMIN LABAREE, The Budd Co.

The Torque Converter and Transmission
—R. M. SCHAEFER, Allison Division, General Motors Corp.

The Diesel Powerplant
—VERNON SCHAFER, JR., Detroit Diesel Engine Division, General Motors Corp.

2:00 p.m.

W. K. SIMPSON, Chairman

Heavy-Duty Oils in Railroad Diesel Engines
—W. E. LASKY, M. A. HANSON, and H. E. FRANK, Gulf, Mobile and Ohio Railroad Co.

Estimating Additive Depletion in Heavy-Duty Oils Used in Railroad Service
—L. A. WENDT, Shell Oil Co.

6:30 p.m.

DINNER

J. A. NELSON, General Chairman of Meeting
T. A. SCHERGER, Chairman, SAE Chicago Section

Toastmaster—ROBERT MORSE, President, Fairbanks, Morse and Co.

J. C. ZEDER, SAE President

"Looking Ahead"
GEN. LEVIN H. CAMPBELL
Executive Vice-President
International Harvester Co.

Friday, Nov. 3

9:00 a.m.

E. E. BRYANT, Chairman

Altitude Effects on Two-Cycle Automotive Diesel Engines
—R. W. GUERNSEY, Detroit Diesel Engine Division, General Motors Corp.

Altitude Performance of the Electro-Motive 567 Engine Under Railroad Conditions
—H. W. BARTH, D. M. LYON and R. B. WALLIS, Electro-Motive Division, General Motors Corp.

1:00 p.m. FRIDAY, NOVEMBER 3

THREE BIG INSPECTION TRIPS

SINCLAIR RESEARCH LABORATORY
Harvey, Illinois

BURLINGTON RAILROAD MAINTENANCE SHOPS, Chicago, Illinois

GM ELECTRO-MOTIVE PLANT
LaGrange, Illinois

NATIONAL MEETINGS

MEETING	DATE	HOTEL
TRANSPORTATION	Oct. 16-18	Statler, New York City
DIESEL ENGINE	Nov. 2-3	Knickerbocker Chicago, Ill.
FUELS and LUBRICANTS	Nov. 9-10	Mayo Tulsa, Oklahoma
	• 1951	
ANNUAL MEETING and Engineering Display	Jan. 8-12	Book-Cadillac, Detroit
PASSENGER CAR, BODY, and MATERIALS	March 6-8	Book-Cadillac, Detroit
AERONAUTIC and AIRCRAFT Engine Display	April 16-18	Statler, New York City
SUMMER	June 3-8	French Lick Springs Hotel, French Lick, Ind.
WEST COAST	Aug. 13-15	Olympic, Seattle, Wash.
TRACTOR	Sept. 11-13	Schroeder, Milwaukee

SAE National

Fuels & Lubricants Meeting

Nov. 9-10, 1950

The Mayo, Tulsa, Okla.

Thursday, Nov. 9

9:00 a.m.

Welcome
—W. K. RANDALL, Chairman, SAE Mid-Continent Section

L. A. McREYNOLDS, Chairman

Performance and Stability of Some Diesel Fuel Ignition and Quality Improvers
—W. E. ROBBINS, R. R. AUDETTE, and N. E. REYNOLDS, III, U. S. Naval Engineering Experiment Station

Liquefied Petroleum Gas as a Fuel for Automotive Vehicles
—LEONARD RAYMOND, Socony-Vacuum Oil Co., Inc.

2:00 p.m.

A. B. BOEHM, Chairman

Small Engines and Dynamometers for Pilot Testing
—W. F. FORD and O. L. SPILMAN, Continental Oil Co.

An Engine Dynamometer Control for Fuel Evaluation by Simulated Road Tests
—A. R. ISITT, M. R. WALL, and A. G. CATTANEO, Shell Development Co.

5:30-6:30 p.m.

Social Hour
Sponsored by
SAE Mid-Continent Section

6:30 p.m.

DINNER

W. J. CARTHAUS, Toastmaster
Program to be Announced

Friday, Nov. 10

9:00 a.m.

J. R. SABINA, Chairman

Antiknock Requirement of Passenger Cars (1949 CRC Octane Number Requirement Survey)
—H. W. BEST, Yale University, J. E. TAYLOR, Gulf Oil Corp., and H. J. GIBSON, Ethyl Corp.

The Potentialities of Fuel Antiknock Quality
—H. E. HESSELBERG and W. G. LOVELL, Ethyl Corp.

2:00 p.m.

HERB RAWDON, Chairman

Fuel and Lubricant Requirements of Personal Type Aircraft Engines
—W. V. HANLEY, Standard Oil Co. of California

Recommendations for Fuel System Design for Personal Aircraft with Regard to Vapor Lock (Report of CFR—AFD of CRC)
—L. L. YORK, Continental Motors Corp., and A. HUNDERE, California Research Corp.

SECTION MEETINGS

British Columbia—Nov. 7

Hotel Georgia, Vancouver, B. C.,—Dinner 6:30—Meeting 7:45. Topic: The New JBS Diesel & DD Fuel Pump. Harold Hall, general service manager, Cummins Engine Co.

Buffalo—Oct. 19

Hotel Sheraton; Dinner 6:30—Meeting 8:00. Topic—Racing Cars vs. Commercial Vehicles—Mauri Rose, engineer Studebaker Corp.

Central Illinois—Oct. 24

Elk's Club, Springfield, Ill. Dinner 6:45—Meeting 8:00. Topic: Development of Gas Turbines—R. C. Allen, mgr. & chief engr., Turbo Powerplant, Allis-Chalmers

Cincinnati—Oct. 15

Plant trip starting at noon to Gulf Oil Co., Hooven, Ohio.

Cleveland—Oct. 9

Wade Park Manor—Dinner 6:30—Meeting 8:00. Topic—Performance Characteristics of Chromium Plated Aircraft & Diesel Cylinders—Russell Pyles, chief engineer, Vander Horst Corp., N. Y.

Dayton—Oct. 10

VanCleve Hotel.—Dinner 6:30 p.m.—Topic: Crosley Car—J. L. Armstrong, vice-president and sales manager, Crosley Motor Corp.

Indiana—Oct. 12

Antlers Hotel, Indianapolis. Dinner 7:00—Meeting 8:00. Topic: Lubrication—W. B. Bassett, Lubrizol Corp. Social hour at 6:00.

Detroit—Oct. 16

Small Auditorium, Rackham Educational Memorial—Meeting 8:00. Technical Meeting—Junior Activity. Topic: Electronic Measuring Instruments—William T. Bean, Director of Industrial Electronics, Inc. A social hour in Snack Bar will follow this meeting.

Detroit—Oct. 23

Dinner Meeting and Plant Tour at Gerity-Michigan Mfg. Co., Adrian, Mich. Tour: 4:30 p.m. Dinner: 6:30 p.m.—Plant tour of USAF Manufacturing Methods Pilot Plant Inspection of Captured German Heavy Press Equipment. Topic: Latest Developments in Heavy Press Manufacture. Speakers: Lt. General K. B. Wolfe,

Continued on Next Page

Deputy Chief of Staff, Material, HQ., USAF—Erwin Loewy, president, Hydropress, Inc.

Metropolitan—Nov. 2

Brass Rail, Fifth Ave. and 43rd St., Dinner 6:30. Meeting 8:00. Topic: Detachable Fuselages. George Lescher of Fairchild. Cocktails and social hour preceding meeting.

Montreal—Oct. 2

Mount Royal Hotel. Dinner Meeting 6:15. Topic: Automatic Transmissions—H. E. Churchill, chief research engineer, Studebaker Corp., South Bend, Ind.

Northern California—Oct. 25

Engineers Club. Dinner 6:30—Meeting 7:30. Topic and Speaker to be announced.

Philadelphia—Oct. 11

Engineers Club—Dinner 6:30—Meeting 7:45. Topic: Gas Turbines for Air and Ground Transportation. A. W. Gabriel, Aviation Gas Turbine Division, Westinghouse Electric Corp.

Pittsburgh—Oct. 24

Webster Hall Hotel—Dinner 6:30—Meeting 8:00. Topic: Engineering at the General Motors Proving Ground—Louis C. Lundstrom, head of Mechanical Engineering Department, G. M. Proving Ground. Special Feature: Colored Films of Proving Ground Activities.

St. Louis—Oct. 10

Congress Hotel. Dinner 6:30—Meeting 8:00. Topic: New Developments in Fuels and Their Applications—H. F. Kelly, assistant manager, Fuel Oil Dept., Socony-Vacuum, N. Y.

Salt Lake—Oct. 30

Hotel Newhouse—Topic: New J.B.S. Diesel Engines and DD Fuel Pump—Harold Hall, general service manager, Cummins Diesel. Special feature: Possibly dinner and film. Picture to follow meeting.

Southern California—Oct. 19

Rodger Young Auditorium—Dinner 6:30—Meeting 8:00. Topic: Adventures in Light Car Design—L. H. Nagler, technical advisor, Nash Motor Div., Nash-Kelvinator Corp.

Spokane-Intermountain—Oct. 20

B. O. F.—Dinner 6:30—Meeting 8:00. Topic: Purpose, Organization & History of SAE—E. W. Rentz Jr., manager, West Coast SAE Office.

Nov. 1

Spokane Hotel—Dinner 6:30—Meeting 8:00. Topic: JBS Diesel Engine & DD Fuel Pump—Harold Hall, general service manager, Cummins Diesel.

Syracuse—Oct. 9

Syracuse Museum of Fine Arts—Dinner 6:30—Meeting 8:00. Topic: Engineering Aspects of Service—C. T. Doman, materials service manager, Ford Motor Co.

Western Michigan—Oct. 17

Sterns Steak House—Dinner 6:00—Meeting 8:00. Topic: Which Fuel—Gasoline? Diesel? Propane? (Symposium). Speakers: L. L. Bowers, Ch. Engr., Waukesha Motor Co., Dick Creger, Continental, F. E. Selin, Indus. Div., Phillips Petroleum

Wichita—Oct. 19

Droll's Grill—Dinner 6:30—Meeting 8:00. Topic: Some Facts on Atomic Energy—Prof. P. S. Albright, Wichita University.

Williamsport—Nov. 6

Dinner 6:30—Meeting 8:00—Topic: Manufacture of Electronic Vacuum Tubes. H. G. Hartwell, plant manager, Sylvania Electric Products, Inc.

Technical Committee Progress

Continued from Page 71

regular oil in a fleet in door-to-door service. The heavy-duty oil reduced crankcase deposits and oil ring groove deposits in both low and high sulfur vehicles.

Other variables that make little difference in engine wear between high and low sulfur fuels, according to the report, are ambient air temperature and geographic location. This conclusion was based on one make of vehicle operated in two fleets in the same type of service, within the atmospheric conditions and temperatures prevailing during the test period. Actual minimum air temperatures were 18 and 45 F.

Only in one fleet did high sulfur gasoline deteriorate the fuel pump diaphragm on the side exposed to the engine crankcase. This wasn't found in low sulfur vehicles or in the other fleets.

Little difference in sulfur effect was reported for a phase of the test program conducted in Canada by the National Research Laboratories, in cooperation with CRC. Military vehicles and drivers were used. Four vehicles of the same make and model were used on each low and high sulfur fuel in each of two tests. One test involved stop-and-go driving during winter; the second involved administrative type service and nonstop runs varying from 15 to 40 miles.

The Canadian tests indicated in up to 1150 miles of cold weather stop-and-go-driving, and up to 2300 miles of mixed driving, there was no significant difference in engine wear between fuels containing about 0.05 to 0.30% sulfur.

The field test methods, sites, equipment, fuels, oils, inspection procedures, and results are fully discussed in the report. The test data and results are documented in detail in charts, tables, and photographs. They should be studied in detail for a correct interpretation of the significance of the tests reported.

The report, CRC-248, has 145 8½ × 11 pages, including 16 charts and photographs and eight tables. It is avail-

able from the SAE Special Publications Department. Price: $3.00 to members, $6.00 to nonmembers.

Tells How to Curb Vapor Lock in Planes

ATTENTION to limitations and critical design features of personal aircraft fuel systems will help minimize vapor lock. These safeguards against vapor lock are spelled out in a recently released CRC report, "Recommendations for Fuel System Design for Personal Aircraft with Regard to Vapor Lock."

An indication of limiting fuel system temperatures for incipient vapor lock are charted in the report. Typical data are given as a function of altitude for fuels of different vapor pressures in the fuel line, carburetor bowl, and diaphragm fuel pump.

Suggestions are made on how to handle the critical design features of both gravity feed and pressure fuel systems. Among those on gravity feed systems are:

1. The gravity tank location should be sufficiently above the carburetor in an airplane to insure adequate operating fuel head to the carburetor at most severe airplane attitude.
2. The fuel selector valve should insure full indexing of all ports; otherwise partially indexed ports can make for restrictions in the fuel system.
3. Fuel lines should be layed out in a smooth, continuous system with the minimum of bends, elbows, and other fittings. Hot areas, such as exhaust stacks, should be avoided. There should be no high spots in the line or accessories to act as vapor traps and thus release vapor in slugs.

A detailed discussion on the fuel pump of pressure fuel systems is given in the report. For example, it advises that submerged tank pumps approach the ideal vapor-lock-free pump. Use of this type pump is recommended as a secondary or auxiliary fuel pump to eliminate many vapor lock problems in current installations. Being submerged in fuel, this pump eliminates suction losses and air leakage on the suction side of the pump. It centrifuges out vapor and air, pumps only liquid fuel.

Last section of the report recommends laboratory and flight test procedures for evaluating vapor-handling capacity of fuel systems. A group under the CFR Aviation Fuels Division of CRC prepared the report.

The report, CRC-247, has 10 $8\frac{1}{2} \times 11$ pages, including three charts. It is available from the SAE Special Publications Department. Price: $.50 to members, $1.00 to nonmembers.

Diesel Fuel Sulfur Gives Varied Effects

NO clear-cut trend on the effect of sulfur in diesel fuels on engine wear and deposits was found by ten laboratories which tested six different fuels, ranging from 0.3 to 1.15% sulfur content, in five type engines. Results of this test program, conducted by the CFR Diesel Fuels Division, are discussed in the CRC Report, "Effect of Sulfur in Diesel Fuels on Engine Operation in the Laboratory."

The test data, notes the report, shows effect of fuel sulfur content on engine deposits and wear to vary with engine type and operating conditions.

One engine type, under high-load operating condition, showed definite correlation between increased wear and deposits with increased sulfur content. Yet the same engine type, operating under low load, showed little or no correlation with sulfur content. This

was also true of other engines under both low and high load operating conditions.

There was no agreement among the various engine types, according to the report.

The researchers who made these investigations conjecture that complex relationships between fuels, engines, and test techniques probably account for much of the wide variations in the test data. Because of the lack of overall consistency plus the variation among engine types, they conclude that:

1. Engine design and operating conditions seem to have a significant effect on wear and deposit reaction to variation in fuel sulfur content.

2. Fuel factors other than sulfur seem to be significant in contributing to engine deposits and wear. Magnitude of this significance is not yet known.

The report recommends further cooperative investigation of the diesel fuel sulfur problem. It suggests emphasis on diesel engine test procedures and techniques to improve engine testing precision, and further study of test fuel selection.

The report also describes the fuels, equipment, and test techniques used; discusses test results in detail; and gives analyses of deposits and used lube oil.

Members of the panel which analyzed the test data were: A. H. Fox, Standard Oil Co. (Ind.), leader; F. C. Burk, Atlantic Refining Co.; H. M. Gadebusch, Detroit Diesel Engine Division, GMC; H. V. Nutt, U. S. Naval Engineering Experimental Station; and L. Raymond, Socony-Vacuum Laboratories.

The laboratories participating in the engine tests were: California Research Corp.; Detroit Diesel Engine Division, GMC; The Pure Oil Co.; Sinclair Refining Co.; Socony-Vacuum Laboratories; Standard Oil Co. (Ind.); Tide Water Associated Oil Co.; U. S. Naval Engineering Experiment Station; Universal Oil Products Co.; and Waukesha Motor Co.

The report, CRC-246, has 119 8½ × 11 pages, including 40 charts and photographs. It is available from the SAE Special Publications Department. Price: $2.00 to members, $4.00 to non-members.

Statistical Quality Control

Continued from Page 68

mental concepts of statistical quality control. (Paper "The Ford Motor Co. and Statistical Quality Control," was presented at SAE Buffalo Section, May 25, 1950. It is available in multilithographed form from SAE Special Publications Department. Price: 25¢ each to members, 50¢ to non-members.)

What Preventive Maintenance Includes

Based on paper by

W. J. CUMMING

White Motor Co.

ALTHOUGH Preventive Maintenance has had a scientific background since the first World War, there is still con-

fusion within the transportation industry as to what it includes. P.M. covers adjusting, tightening, lubricating and cleaning and is distinct from repair. It does not eliminate repair, but does postpone repairs until parts and assemblies have served their full life expectancy.

All truck P.M. schedules must contain the same inspection procedures or identical checking processes. The only important difference in schedules is in when to inspect and at what life expectancy period action must be taken.

Studies indicate there should be approximately 70 operations in every complete P.M. program, regardless of the vehicle. These can be separated into five groups, all the items in each group having a comparable life expectancy.

In city transportation or in truck operation up to 25,000 miles annually, traffic conditions are found to give long periods of engine idling and to cause excessive wear of clutch and brake, hence the wear of these three units in terms of mileage is in the ration of three times the road or tire mileage. For trucks having an annual mileage of 25,000 to 100,000 and more, the vehicles wear is in direct proportion to the road or tire miles. (Paper "The Origin of Preventive Maintenance and What It Includes," was presented at SAE New England Section, February 7, 1950. It is available in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to non-members.)

Aspects of Military Aircraft Standardization

Based on paper by

R. C. RETHMEL

Air Materiel Command

THIS paper discusses military aeronautical standardization from four angles: 1. Governmental standardization organizations and procedures. 2. Objectives of standardization. 3. Military policies as affecting supply and maintenance, procurement, inspection, production and cooperation. 4. Future needs and responsibility of the Military Service.

(Paper "Military Aeronautical Standardization," was presented at SAE National Aeronautic Spring Meeting, New York, April 17-20, 1950. It is available in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to non-members.)

Owners Rate Qualities Desired in Upholstery

Based on paper by

W. F. BIRD and C. L. CONLEY

Collins & Aikman Corp.

CAR owners know next to nothing about upholstery. With the exception of those who own nylon trimmed cars, they don't even know what kind of upholstery is in their cars. They do know what qualities they think most desirable.

They rank durability first and fore-

most. Then, in the order of importance: ease of cleaning, softness and smoothness of material, and color. In every survey made, durability came first, indicating a primary concern with the practical features of upholstery. Color and appearance, which are matters of style, were rated well up, but dealers put more emphasis on style than do the owners.

After being told that some fabrics are easier on clothing than others, and that some are safer, owners gave a higher rating to these qualities and the ranking then became: durability, ease on clothing, ease of cleaning, safety, and softness and smoothness of material. In general, safety was given a higher rating by those who had children under sixteen than by the childless.

Durability was a factor in car reputation even before the last war. Cars with durable, easily cleaned upholstery sold rapidly as used cars. Buyers tended to judge a car's condition by the state of the upholstery, and dealers saved money on reconditioning.

In a survey made by duPont, owners were found to be well pleased with nylon pile and cord combinations, but to be dissatisfied with nylon slip covers. Half the owners found such covers difficult to clean and too slippery.

(Paper, "Development of Modern Upholstery Fabrics," was presented at SAE Summer Meeting, French Lick, June 4-9, 1950. It is available in multilithographed form from SAE Special Publications Department. Price: 25¢ each to members, 50¢ to nonmembers.)

Alcohol-Water Injection Discussed

Based on paper by

JAMES C. PORTER

Northern Regional Research Laboratory

HIGHER compression ratios require higher octane fuels, yet most vehicles, with the possible exception of heavy trucks, use the full octane value of their fuel for only a fraction of the driving time.

Over most of the part-throttle range, when the engine is operating essentially at a lower compression ratio, maximum octane requirements are not needed. Even for the high compression ratio like the 10:1 of the General Motors test engine, where the peak octane requirement is slightly more than 100, premium grade gasoline is adequate.

Since the use of higher octane fuels to give knock-free performance over the entire range of local conditions results in loss of valuable octane numbers, it is suggested that alcohol-water injections be used. They satisfy the engine's demand for better fuels without increasing octane numbers.

The alcohol-water mixture is automatically injected when needed by an auxiliary carburetor device controlled by manifold vacuum. The efficiency of such mixtures to suppress knock is about the same for ethanol and methanol and the addition of 15% water appears to make very little difference in the amount needed.

(Paper "Alcohol-Water Injection for High Compression Engines," was presented at Bradley University Student Club of SAE, Peoria, March 7, 1950. This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

Turn to Page 92

Maintenance Demands Greater Accessibility

Based on paper by

ROBERT CASS

White Motor Co.

TO meet the need for lower maintenance costs, design engineers have tried to improve a part to make it last longer in service and hoped that intervals between removal or repair would be so long as to make inaccessibility relatively unimportant.

Increasing part durability is inadequate today with the high cost of labor. Accessibility must now be given primary consideration because maintenance costs can make the difference between profit and loss in operation.

Responsibility rests with the design engineer to find the best balance between the original cost of the part and the economic consequences of the high labor cost for removal and replacement. It is his responsibility, too, to see that maintenance hours on new models show a reduction over those of previous models, regardless of what is demanded in the way of styling and so-called streamlining by the sales department and operators. (Paper "Accessibility," was presented at SAE National West Coast Meeting, Los Angeles, August 14-16, 1950. It is available in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to non-members.)

Diesel Economy Stems From Higher Btu Usage

Based on paper by

MERRILL C. HORINE

Mack Mfg. Co.

THE diesel begins to show a saving over the gasoline engine when annual fuel expense exceeds twice the interest and amortization on the additional price of the diesel-powered vehicle. Higher thermal efficiency, rather than difference in price between gasoline and diesel fuels alone, accounts for these economies.

For example, assume a gasoline-powered vehicle giving 4.5 mpg and operating 60,000 miles annually. At 21¢ per gal, annual fuel costs for 13,333 gal will be $2800. Since diesel fuel economy savings are consistently about 50%, the diesel will save $1400 per year.

Now take a truck like this one, equipped with a diesel engine of equal power, priced $1700 above the gasoline truck. Fuel savings will offset this entire additional investment in less than 15 months. After that, the fuel savings remain.

But this is not a clear gain. On the additional $1700 price will be an annual investment charge of say 6% interest on the average amortized investment, or $51, and an amortization of the extra price at the rate of 300,000 miles for five years.

This amounts to $340 per year, making a total investment charge of $391 per year. The entire extra investment, at its full 6% interest, will be offset by fuel savings in little more than 15 months. After that the $1400 will be net gain. On an original investment of $1700, plus $340 amortization, or $2040, this amounts to 68% return.

Some feel the increase in vehicle weight due to the diesel installation is not reflected in such a comparison. Only possible penalty is effect of increased tare weight, since effect of this additional weight on fuel con-

Wagner Air Brakes
THE CHOICE OF FLEET OPERATORS

The statements at right—by men who are responsible for the economical operation of heavy-duty vehicles, are just a few examples of what hundreds of cost-conscious operators have said about Wagner Air Brakes.

In the trucking industry everywhere—on all types of heavy-duty vehicles, Wagner Air Brakes are establishing outstanding economy records for brake performance under the most severe operating conditions. Because of their proven reliability, Wagner Air Brakes have become the choice of fleet operators.

Wagner Air Brakes are the product of more than twenty years of brake engineering experience—experience gained in the manufacture of hydraulic brakes and brake parts for the automotive industry. This outstanding brake knowledge is your assurance that when you install Wagner Air Brakes you will reduce brake maintenance costs and increase your payload profits.

"FOR FREEDOM FROM MAINTENANCE... FOR ALL AROUND DEPENDABILITY... WAGNER AIR BRAKES ARE OUR BEST BUY."

B. E. Garner, *Maintenance Supt.*
WESTERN TRUCKING COMPANY

"ONE OF THE FEATURES THAT SOLD US WAGNER AIR BRAKES IS THE ROTARY AIR COMPRESSOR."

F. Harrison, *Garage Foreman*
NIGHTHAWK FREIGHT SERVICE

"OUR FLEET OF HEAVY-DUTY ASPHALT HAULERS REALLY GETS A GOING OVER—WE MADE THE RIGHT CHOICE WHEN WE SELECTED WAGNER AIR BRAKES."

F. L. Hunter, *Plant Superintendent*
MISSOURI PETROLEUM PRODUCTS CO.

SEND FOR THIS BULLETIN

Everyone responsible for the economical operation of heavy-duty vehicles should have Wagner Bulletin KU-50. A request will bring your copy by return mail.

Wagner Electric Corporation
6378 Plymouth Ave., St. Louis 14, Mo., U. S. A.

LOCKHEED HYDRAULIC BRAKE PARTS and FLUID...NoRoL...
CoMaX BRAKE LINING...AIR BRAKES...TACHOGRAPHS...
ELECTRIC MOTORS...TRANSFORMERS...INDUSTRIAL BRAKES

sumption has already been allowed in the saving of 50% of the gasoline bill.

Whether it is a penalty depends on legal weight limitations in states in which the truck operates. If the vehicle is up to top weight with the gasoline engine, and maximum weight limitations or the Bridge Formula are in force, this additional weight decreases maximum payload. The most weight conscious operators, those on the Pacific Coast, value payload capacity at $1.00 per lb per year. Here is how this affects the picture in one particular case:

A popular diesel vehicle has an additional weight of 573 lb. From our $1009 net fuel cost savings we must subtract $573 (assuming the $1.00 per lb per year for lost payload). This leaves a $436 net saving. At 41,314 miles per year fuel savings exactly balance the sum of the investment charges plus payload penalty. This is the minimum economical annual mileage in this case.

Here is how it is figured:

$$\text{Minimum Economical Annual Mileage} = \frac{2\,(0.23 \times \text{Extra Price}) + \text{Extra Weight, lb}}{\text{Gasoline Price per gal}} \times \text{Gas mpg}$$

In states where axle weight limitations govern, there is no weight penalty since weight on the rear axle is unaffected. Minimum economical mileage in the case cited would go down to 16,658.

In these states, the formula is:

$$\text{Minimum Economical Annual Mileage} = \frac{2\,(0.23 \times \text{Extra Price})}{\text{Gasoline Price per gal}} \times \text{Gas mpg}$$

Applying this test to every truck installation would indicate that many vehicles could economically be replaced with, or changed over to, diesel equipment. The old credo that diesels are economical only in big-mileage operations appears questionable.

While city trucks on short hauls cover less mileage than highway freighters, their miles per gallon with gasoline also is apt to be less. Thus the diesel fuel savings per mile are likely to be greater. Big diesel virtue is its high economy at part load and idling.

How does the diesel achieve these economies? Certainly not merely by the price differential between gasoline and diesel fuel. Diesels have proved economical even when their fuel costs more than gasoline. Real source of diesel economy is higher thermal efficiency, which produces about 75% more miles per gallon. This is a 43% saving in itself.

Diesels convert a greater percentage of Btu's in fuel to active power than gasoline. A steam locomotive shows about 5% thermal efficiency; gasoline engines about 25%; and diesels, about 35%.

This higher thermal efficiency stems from several things inherent in the diesel . . . higher compression pressure, higher average air-fuel ratios, and higher average volumetric efficiency.

Higher compression pressures mean greater specific power output and lower specific fuel consumption. Higher compression ratios in gasoline engines call for more costly higher-octane gasoline and bring problems.

Air-fuel ratios in gasoline engines vary but little between 13 and 14 to 1 by weight. Those of the diesel vary from 20 to 1 at full load up to as much as 133 to 1 at idle. Modern diesels

It's not the container size that counts...

it's the filtering surface!

That's why the accordion element in Purolator's Micronic* Oil Filter removes 290% more abrasives...

PERCENTAGES BY WHICH PUROLATOR MICRONIC ELEMENT EXCELLED COMPETITIVE TYPES

COMPETITOR	IN AVERAGE DIRT RETENTION PUROLATOR LED BY:
A	199%
B	220%
C	113%
D	547%
E	164%
F	619%
G	255%
H	339%
I	318%
J	193%
K	237%

AVERAGE PUROLATOR SUPERIORITY **290%**

Just look! This smallest automotive-type Purolator Micronic* element has 570 square inches of effective filtering surface compared to 54 square inches in ordinary filters! Yet it's no larger in container size.

Purolator engineers perfected this element so that it filters particles measured in microns (.000039 of an inch) . . . removes an average of 290% more abrasives as proved by competitive test results shown above.

These Purolator advantages add up to faster, more complete filtering of all the sludge and abrasives . . . longer engine life with fewer repairs . . . greater acceptance for your engines and vehicles equipped with Purolator. Greater acceptance because car owners everywhere learn about Purolator through the most aggressive advertising campaign in the oil filter industry!

If you have a special filtering problem, let our technical staff lend a hand. They're the most experienced filter engineers in the business!

*Reg. U. S. Pat. Off.

PUROLATOR

MICRONIC OIL FILTER

"FIRST IN THE FIELD OF FILTERING"

PUROLATOR PRODUCTS INC.
Rahway, New Jersey and Toronto, Ontario, Canada

consume no fuel in drifting or decelerating. Gasoline engines continue to consume fuel as long as they turn.

Fuel injection and compression ignition combine to yield these superior characteristics. Since it takes in pure air only through the intake manifold, without throttling and without mixing with the fuel, the diesel operates at full compression at all times. It breathes easier than the gasoline engine, since high air velocity through the carburetor for atomization is not needed. Distribution between cylinders is better than that possible in the gasoline engine because of lower velocity and freer valve porting. (Paper "Application of Diesel Engines to Road Vehicles," was presented at SAE Texas Section, Houston, Feb. 24, 1950, and at SAE Pittsburgh Section, March 28, 1950.)

Fuel Economy High In '50 Stock Car Run

Based on paper by

MAX EPPS

General Petroleum Corp.

and W. S. MOUNT

Socony-Vacuum Oil Co.

IN the 1950 Mobilgas Grand Canyon Run, the competing 1950 stock-model, four-door sedans averaged 22.07 mpg and 50.29 ton-miles per gal. (The ton-miles per gallon is the highest of the seven runs of the series.)

The cars were stock models, conforming to factory specifications. They were driven in a normal way, with all "stop" signs and legal speed limits observed. The cars averaged 41.47 mph.

The run was made in two days. The course covered 751.3 miles and ran from the center of Los Angeles, north over the mountains, across the desert to Lone Pine, Calif., and east across Death Valley to Las Vegas. It covered an altitude range from 178 ft below sea level to 7005 ft above sea level, and a temperature span of 28 to 68 F.

Results of the run emphasize the high mileage level available from 1950 cars and gasoline by adhering to safe driving methods and good servicing practices. (Paper "1950 Mobilgas Grand Canyon Run," was presented at SAE Northwest Section, Seattle, March 10, 1950. It is based on work done by Wilmot Sandham, General Petroleum Corp., and E. J. Sanders, consulting engineer. This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

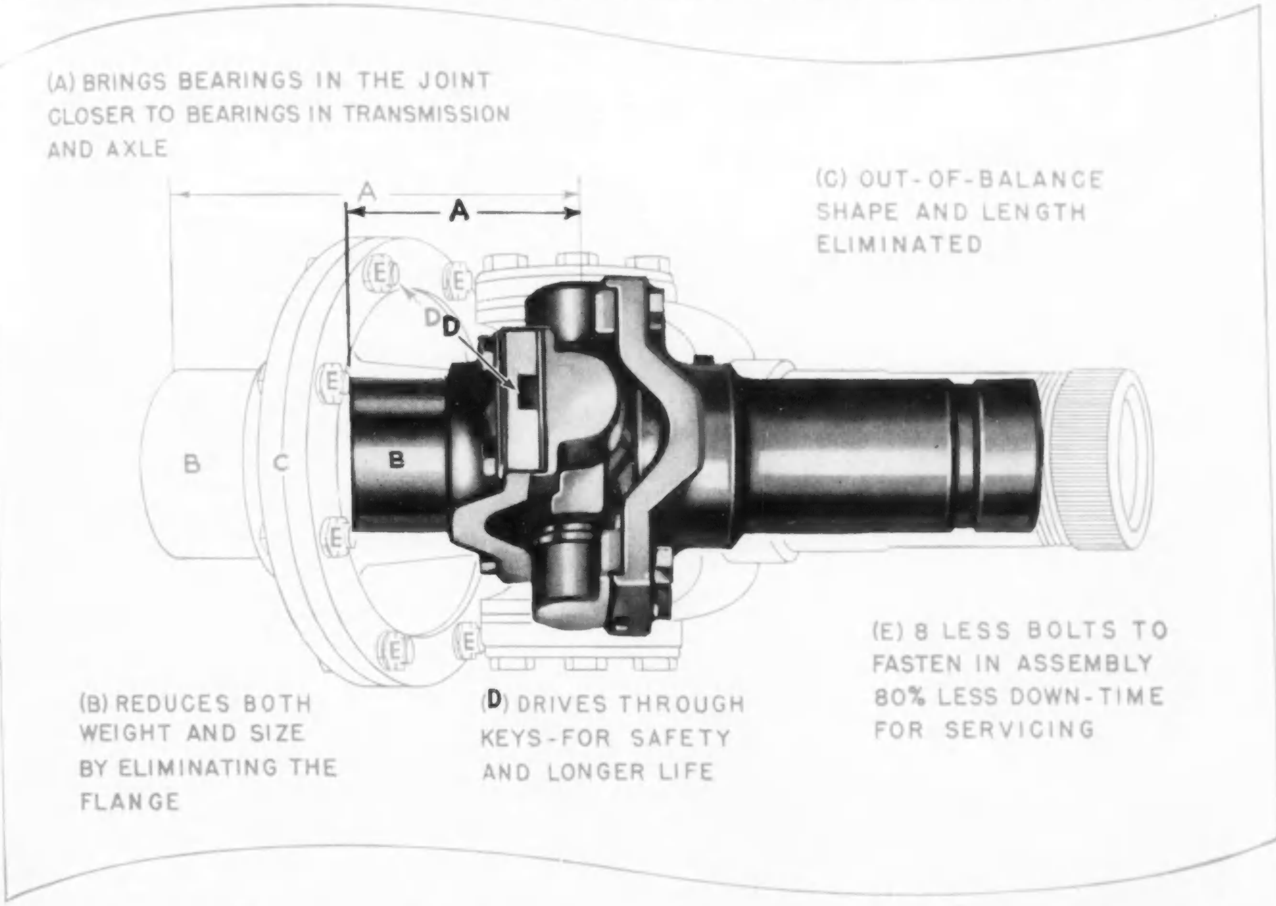
About SAE Members

Continued from Page 76

BARCLAY F. SMITH is now employed by the Precision Automotive Components Co., St. Louis, Mo., as purchasing agent. Prior to this, he was an experimental engineer with the Carter Carburetor Corp.

THEODORE T. CORRELL, who previously was service manager with Hubach & Parkinson Motors, Oregon City, Oreg., is now fleet superintendent

ENGINEERED *To Give* YOUR TRUCK 5000 TON-MILES *More* PAYLOAD



By reducing *deadweight* 28 pounds — through using MECHANICS 34% lighter UNIVERSAL JOINTS — a truck can increase its PAYLOAD over 5,000 ton-miles (during 400,000 miles of operation) *without extra cost*, or loss of capacity. The high-strength-to-weight ratio, 34% lighter MECHANICS Roller Bearing UNIVERSAL JOINTS Truck PROPELLER SHAFT places less load on the transmission and pinion bearings, and runs smoother. Let our engineers help add to the ton-mile PAYLOAD capacity of YOUR trucks, by specifying a weight-saving MECHANICS Roller Bearing UNIVERSAL JOINTS Truck PROPELLER SHAFT application.

MECHANICS UNIVERSAL JOINT DIVISION
Borg-Warner • 2022 Harrison Avenue, Rockford, Illinois

MECHANICS
Roller Bearing
UNIVERSAL JOINTS
For Cars • Trucks • Busses and Industrial Equipment

with Risberg's Rand Truck Line, Inc., Portland, Oreg.

FRANK B. ROBB, formerly a general contractor in Glendora, Calif., is now part owner of Robb & Robb, Cleveland, Ohio. His new work entails the preparation and prosecution of patent and trade mark applications.

CHARLES L. FREEL has been appointed manager of the new field engineering department at the Lord Mfg. Co., Erie, Pa. Prior to this, Freel was sales engineer for the same company.

ROBERT J. LEWARK is now with the Marvel-Schebler Carbureter Division of the Borg-Warner Corp., Decatur, Ill. Prior to this, he was a refrigeration engineer with the American Central Division, Avco Mfg. Corp., Connorsville, Ind.

LONNIE J. THOMAS is now an insurance agent with the College Life Insurance Co. of America, Indianapolis, Ind. Prior to this, he was a sales engineer with Redmond Co., Inc., Owosso, Mich.

JULIAN SOUKEY, formerly a test engineer with Wright Aeronautical Corp., Woodridge, N. J., is now a trainee in the college graduate training program of the Caterpillar Tractor Co., Peoria, Ill.

PAUL PHILLIPS is employed by Cummins Diesel Sales of Washington, Seattle, Wash., as shop superintendent. The company is engaged in the sales and maintenance of Cummins Diesel Engines. Previous to this, Phillips was a diesel engineer with Automotive Products Co., Portland, Oreg.

JOHN R. ANTHONY, JR., who, prior to this, was a sales engineer in the Washington, D. C. office of the New Departure Division of GMC, Bristol, Conn., is now a student at the University of Texas Law School, Austin, Texas.

CLIFFORD LACE is employed as a draftsman with the Electric Controller & Mfg. Co., Cleveland, Ohio. He was formerly employed in a similar capacity by Jack & Heintz Precision Industries, Inc., Cleveland, Ohio.

DONALD C. FISH, who was plant superintendent for the Trailmobile Co., is now a tool designer (checker) with Chance Vought Aircraft in Grand Prairie, Texas.

LOREN E. LURA is now a bearing application engineer with Hyatt Bearings Division of GMC in Chicago.

THOMAS H. McCONNELL, JR., has been appointed sales promotion and advertising manager of Vulcan Rubber Products, Inc., Brooklyn, N. Y., in charge of all new business. McConnell joined the company in 1948. He was formerly with the Standard Oil Co., N. J., and the Standard Vacuum Oil Co., where he traveled extensively in Europe, Africa and the Orient as an aviation representative.

EDWARD F. OBERT, associate professor of the mechanical engineering department at Northwestern University has written a new 590-page textbook titled, "Internal Combustion Engines." The book is being published by the International Textbook Co., Scranton, Pa., and will be available for instruction use this fall. In 1949 Professor Obert completed a textbook on, "Elements of Thermodynamics and Heat Transfer." A year earlier, he was the author of another textbook titled, "Thermodynamics." These books were published by the McGraw-Hill Book Co., N. Y.

WILLIAM G. NEWMAN, formerly chief engineer with Sherman Products,

Inc., Royal Oak, Mich., is now connected with Jered Engineering Co., Detroit, as project engineer. His new position entails leadership in the development and research of vehicles for the Bureau of Ships.

OLIVER W. INSKEEP is no longer with the Dalmo Victor Co., San Carlos, Calif., where he was in charge of the mechanical testing laboratory. He is now self employed as a consulting engineer in Detroit, Mich.

W. WELSH GODON is now employed in the capacity of power plant group engineer by the Piasecki Helicopter Corp., Morton, Pa. His new position entails the supervision of the design of helicopter power plant installation. Prior to this, he was employed by the Republic Aviation Corp., Farmingdale, N. Y., in a similar capacity.

DAVID E. BASOR, formerly shop superintendent with Murphy White Trucks, Seattle, Wash., is now with Larson Construction, Astoria, Oreg., in the capacity of master mechanic.

WILLIAM H. CHAPMAN has been appointed director of engineering to coordinate machine product design research and application engineering at Hyatt Bearings Div., GMC, Harrison, N. J. Chapman joined Hyatt in 1922 as a foreman in charge of race grinding. He has had varied engineering and manufacturing experience, having been supervisor of development, methods and equipment, superintendent of dual purpose bearing manufacturing, and assistant factory manager in charge of engineering and development, from which position he has been promoted to his present status. During World War II he handled all of the Division's contracts with government agencies.

SIMON DE SOTO is now employed as a research engineer with the Stratos Corp., Division of Fairchild Engine & Airplane Corp., Farmingdale, N. Y. He is engaged in high speed turbine research. Prior to this, he was an instructor in the mechanical engineering department of Syracuse University, Syracuse, N. Y.

RONALD ROSS, who, previous to this, was a body layout-man with General American Aerocoach, East Chicago, Ind., now holds a similar position with Studebaker Corp., South Bend, Ind.

RIVINGTON STONE, formerly chief of drafting with AMATC, Eglin Air Force Base, Valparaiso, Fla., is now a mechanical engineer with the Maryland Electronic Mfg. Corp., College Park, Md. The company produces electronic equipment and Stone is responsible for the mechanical design of it.

WALTER W. KADOW, previously truck sales manager with Josceylyn Motors Corp., Culver City, Calif., is now connected with Kadow's White Truck Sales, Grand Rapids, Mich., in the capacity of general service manager.

DAVID H. KAPLAN has resigned his position as project engineer with Doman Helicopter, Danbury, Conn., to become president of Convertawings, Inc., New York. Convertawings is an engineering organization devoted to the design and development of convertible aircraft.

WILSON P. GREEN, professor of mechanical engineering at Illinois Institute of Technology, became assistant chairman of the Armour Research Foundation's applied mechanics department in charge of engine and lubricants research. Green received his bachelor's and master's degrees from the University of Florida. He was an engineer with the U. S. Department of Agriculture from 1936 to 1938, and from 1938 to 1946 was associate professor of mechanical engineering at the University of Maryland. He joined Illinois Tech in 1946.


"A USED TRUCK I BOUGHT LAST YEAR ALREADY HAD BENDIX-WESTINGHOUSE AIR BRAKES . . .
Savings on that ONE Truck CONVINCED ME!"
Running a small fleet of trucks calls for some mighty close figuring if you are going to realize the maximum profit on every job. That's why old hands in the trucking business install Bendix-Westinghouse Air Brakes—they *know* they can depend on extra savings. When you total up those savings in maintenance and parts replacement costs, plus the value of the added time on the job, you realize what a good business investment Bendix-Westinghouse Air Brakes can be. Drivers prefer them, too, because the added confidence and reduced physical and mental strain mean better road time. Whether for old or new trucks, be sure of the best – always specify Bendix-Westinghouse Air Brakes.
THE BEST BRAKE IS
AIR
BENDIX-WESTINGHOUSE
GENUINE
AIR BRAKES
REG. U.S. PAT. OFF.
THE BEST AIR BRAKE IS
Bendix-Westinghouse
BENDIX-WESTINGHOUSE AUTOMOTIVE AIR BRAKE COMPANY
ELYRIA, OHIO

THOMAS J. BURKE, formerly a service manager with Rollstone Buick, Inc., Fitchburg, Mass., now holds a similar position with New London Buick, Inc., New London, Conn.

VALENTINE F. PABST is now employed by the Arrow Machine Co., Newark, N. J., in the capacity of field representative. His new position entails the soliciting of engine rebuilding in the truck and bus field, and the handling of all service calls. Prior to this, he was regional representative in the piston rings department of Koppers Co., Inc., Baltimore, Md.

ROBERT S. CRAIG, formerly general manager of Territorial Motors, Ltd., Honolulu, Hawaii, has been promoted to director. He is now chairman of the executive committee which entails assisting the management and directors in conducting the business. Craig is also operating as a private consultant to the Hawaiian Economic Foundation, the Industrial Research Advisory Council, and others.

WILLIAM J. CRAWFORD III is now employed by General Electric Co., Lynn, Mass., in the capacity of technical engineer. Prior to this, he was an engineer with the Chrysler Corp., Highland Park, Mich.

LEONARD G. SCHNEIDER, formerly a service engineer with the Pontiac Motor Division of GMC, Pontiac, Mich., is now a consultant with F. P. Willcox, Washington, D. C. The company produces precision mechanical and optical equipment.

ARTHUR C. BATES, who, prior to this, was a professor of mechanical engineering at Lehigh University, Bethlehem, Pa., is now manager of the development and design section of the engineering department of the Railway & Industrial Engineering Co., Greensburg, Pa. He is responsible for the development and final production design of all new products and major redesigns of existing products.

J. PERRY BARKER will be in charge of the new offices in Mineola, N. Y. of the Adel Division, General Metals Corp., manufacturers of hydraulic valve equipment for both the aviation and industrial fields. At the same time, he will continue to act as Adel's eastern representative.

NORMAN C. MILLMAN is now chairman of the planning board in the city of Oshawa, Ontario. Prior to this he was engaged in service promotion with General Motors of Canada, Ltd., Oshawa, Ont.

NORMAN DAMON, vice-president of the Automotive Safety Foundation, is featured in "LOOK Applauds" in the September 26th issue of Look. "LOOK Applauds" is devoted to giving recognition to outstanding personages who have made distinguished contributions

to knowledge, culture, and the improvement of human relations.

GEORGE W. WUNNER has been appointed manager of the sales and installation engineering department in the Propeller Division, Curtiss-Wright Corp., Caldwell, N. J. Wunner was promoted to his new post from the position of assistant manager of the department. He originally joined the Propeller Division in May, 1941. Since May, 1943, he has been a member of the sales department staff, responsible for engineering coordination, sales promotion, and customer relationships with major aircraft manufacturers.

ALLEN TAYLOR has completed his 20th year of service with Shell Oil Co., New York. A veteran of both world wars, Taylor joined Shell in 1930 as a technical inspector in the company's marketing-operations department. He is co-inventor of Shell's patented 'eductor system,' a part of the valving equipment and one of the main features of the latest type aviation refuelers.

ALVIN P. WILLIAMS, JR., is now a technical writer in the Service Engineering Division of the Beech Aircraft Corp., Wichita, Kans.

New Members Qualified

These applicants qualified for admission to the Society between Aug. 10, 1950 and Sept. 10, 1950. Grades of membership are: (M) Member; (A) Associate; (J) Junior; (SM) Service Member; (FM) Foreign Member.

Atlanta Group

Carl R. Allen (M).

Baltimore Section

Major Walter C. Featherston (A), Harold E. King (A).

Canadian Section

Robert Meyric Ellis (M), Robert John Whitla (A).

Central Illinois Section

Edward Daniels Grier (J), Jacob S. Smith (A).

Chicago Section

Walter William Frank (M), Nicholas L. Heinz (A), M. G. McGregor (A),

Turn to Page 107

G. Mortensen (A), Thur L. Schmidt (M).

Cleveland Section

Hans E. Fueger (M), Leonard Patrick Kane (J).

Dayton Section

William B. Boyd (J).

Detroit Section

Newell P. Beckwith (M), John S. Bertling (A), Norman J. Downey (J), John W. Drake (A), David J. Dunlop (M), E. L. Harrig (M), Robert D. Kemp (A), Lester W. Klouser (M), Harold E. Lowe (M), J. R. Mahan (M), Vernon E. Riddell (M), Stanley Smolen (A), Andy Torok (M).

Kansas City Section

Richard Henry Sauer (J).

Metropolitan Section

Al Anderson (M), Peter A. Cipriano (A), Albert Malek (M), Lars-Georg Romberg (M).

Mid-Continent Section

Eulan C. Kortge (J).

Milwaukee Section

Haskell M. Reichert (J), Paul Gerard Willer, Jr. (J).

Montreal Section

Frank Blashford Thompson (A).

New England Section

John William Jacobsen (J).

Oregon Section

Rowland L. Miller, (A).

Pittsburgh Section

William J. Kysar (A).

Southern California Section

George G. Bell (A), John Albert Meursinge (J), Wilmot Sandham (M), Paul W. Sheehan (M), True R. Slocum, Jr. (M).

Southern New England Section

Herman Dvorak (J).

Syracuse Section

V. A. MacMillan (M).

Texas Section

W. F. Kascal (A), Edwin L. Stoorza, Sr. (A).

Outside of Section Territory

Edward D. Badey (M), William L. Cook (J), Frank DeFrance (A), Raymond Henry Eppling (J), Millard Arnon Pinney (M), A. R. Vickers (A).

Foreign

Alfredo I. Christlieb (M), Mexico.

Applications Received

The applications for membership received between Aug. 10, 1950 and Sept. 10, 1950 are listed below.

British Columbia Group

William Fairhead.

Canadian Section

Alfred Anthony Alessio, William Close, James William Fraser, Bert Hannaford, Sidney L. Kent.

Chicago Section

Donald L. Derebey, Donald H. Fidler, Paul T. Hahn, John W. Kuczwara, Raymond Lampsa, John S. Marshall, Jr., Donald L. Powell, Robert William Travis, Donald William Wanderer.

Cincinnati Section

James O. Chaskel.

Cleveland Section

Arlie Lyman Brown, Frank Kreiner, John R. Lowe, Jr., Elmer J. Scheutzow, Charles Stack, Raymond J. Stewart, Arthur G. Wahrenberger.

Colorado Group

David R. Vondy.

Dayton Section

Marvin R. Davis.

Detroit Section

Ellsworth C. Adams, Henry S. Budden, Nicholas P. Christy, William Corley, Douglas Dow, Michael Durella, Robert H. Eaton, Richard Cullen Edwards, Robert Warren Fogerty, Jr., Victor Francis, Lewis Kinmple Garis, Jr., David A. Gorte, Robert W. Graham, John B. Harrison, Elmer C. Lang, Theophil M. R. Lupfer, Henry J. Malik, Jr., Jess Marosi, Charles Louis Payor, George Rupinski, John Gallus Schaub, Jr., Donald B. Smith, Aram Sogoian, John F. Zimmerly.

Hawaii Section

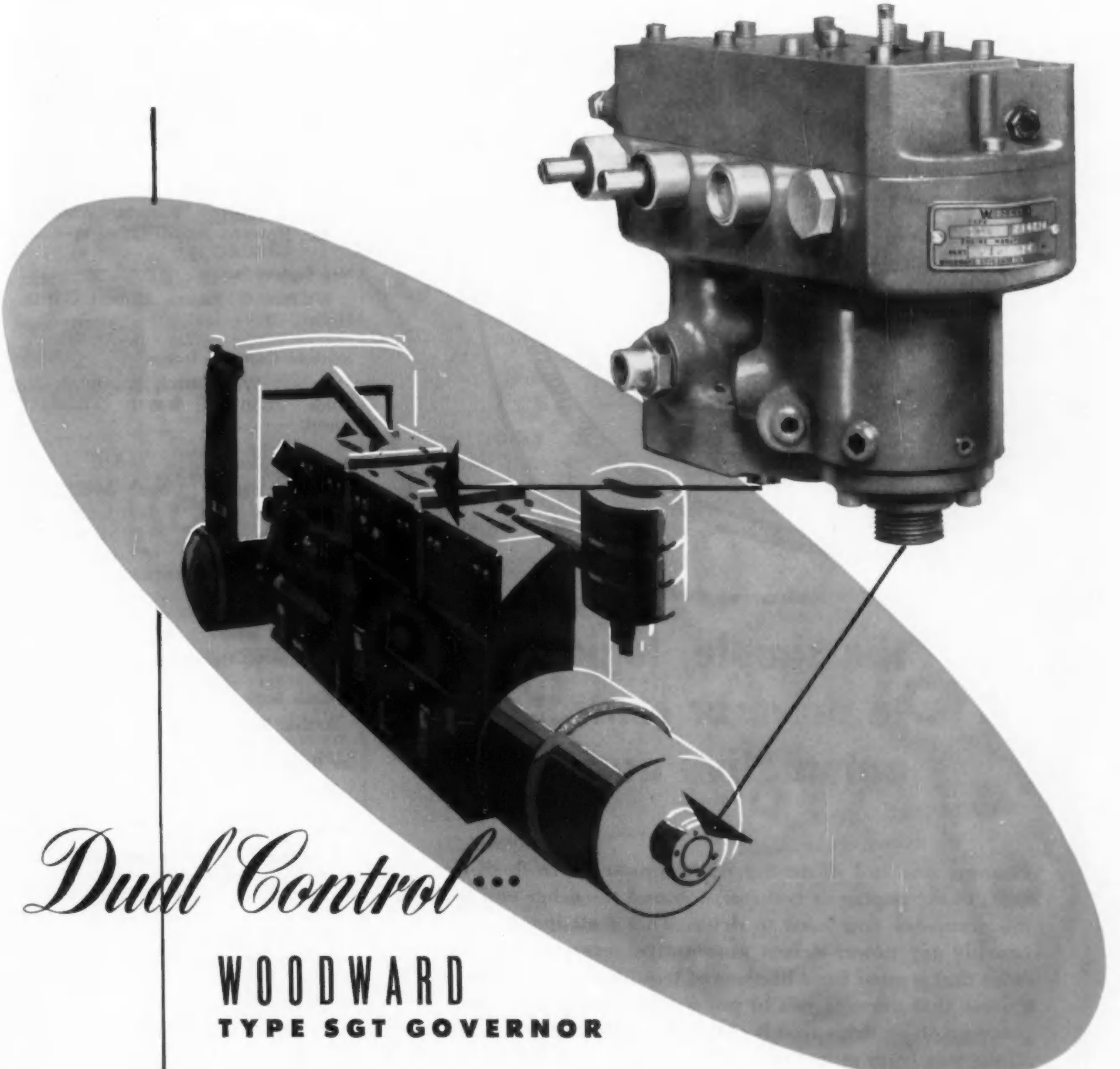
William Henry Maguire, Harry S. Mizuta.

Indiana Section

Alan S. Clark, Edward Gayle Bulpitt, Rowland M. Hussey, Jr., Hubert B. Irwin, John H. Sears.

Metropolitan Section

Herbert Aronson, Eugene Leonard Brickman, Fred O. Dietsch, Jr., Eberhard Dullberg, William Fassuliotis, Walter L. Hermes, Warren F. Menicke, Harry Tennent Rittenhouse, Edward A. Sammis, Richard John Scanlan, Jr., Walter E. Thielhart.



Dual Control...

WOODWARD

TYPE SGT GOVERNOR

A single governing unit provides limitation of engine and converter shaft speeds, governed control of engine power level to maintain desired output shaft speed with varying loads, and engine power level control to vary output shaft speed down to stall.

World's oldest and largest exclusive manufacturer of hydraulic governors for prime movers.

WOODWARD GOVERNOR COMPANY

ROCKFORD, ILLINOIS

Mid-Continent Section

Paul Heinig, Jr., Thomas Robert Streets.

Milwaukee Section

Paul M. Beard, John E. Heuser, Clarence LeRoy Kirsch, Walter H. Losse, Howard Floyd Traeder.

New England Section

William O. Faxon, Hubert Gardiner Hoben, Jr.

Northern California Section

Robert W. O'Hara, Hardy G. Reynolds, John K. Smyth, Thomas J. Turek.

Northwest Section

Nevin Harvey Cope, Averil Dean Walker.

Oregon Section

James Eugene Doeneka, Lester Earl Kassebaum.

Philadelphia Section

Arthur M. Berg, Edwin R. Berg.

St. Louis Section

Arthur Bruce Cook, Harvey D. Ferris.

San Diego Section

Henry K. Hauser.

Southern California Section

Clarence T. Rasmussen, Edward M. Rose, Clarence E. Siemonsma, Allen L. Simms, David E. Westerling.

Southern New England Section

George B. Schwartz.

Syracuse Section

Dean Clarence Broughton.

Twin City Section

Wendell E. Bergren, Lee C. Paulson.

Virginia Section

Ralph Allen Amos, Beverly R. Belcher.

Washington Section

Charles E. Greeley, John L. Lackler.

Western Michigan Section

Charles E. Bluhm, Jack M. Hamilton, D. M. Hesling.

Williamsport Group

George Everett Billman.

Texas Section

Roy L. Adams, A. G. Colburn, Jr., Jerry Kulick, Wix C. Thorpe.

Outside of Section Territory

Joseph A. Bennett, Richard L. Catlin, Thomas Homer Evans, Walter Blake

King, Jr., Robert Sherman, Warren H. Smith.

Foreign

William John Collins, Brazil, S.A.; Henry M. Haan, Belgium; M. Sreenivasa Rao, India.

HAVE YOU Changed Your Address?

So that your SAE mail will reach you with the least possible delay, please keep SAE Headquarters and the Secretary of your local Section or Group advised of any changes in your address. Such notices should be sent to:

1. Society of Automotive Engineers, Inc., 29 West 39th St., New York 18, N. Y.
2. The Secretary or Assistant Secretary of your Section or Group at the addresses listed below:

Baltimore

R. L. Ashley, Ashley Chevrolet Sales, Inc., 2001 N. B'way, Baltimore 10, Md.

British Columbia

John B. Tompkins, British Columbia Section, SAE, 1010 Dominion Building, 207 W. Hastings St., Vancouver, B. C., Canada

Buffalo

C. J. Lane, 1807 Elmwood Ave., Buffalo 7, N. Y.

Canadian

F. G. King, Maclean-Hunter Publishing Co., Ltd., 481 University Ave., Toronto, 2, Ont., Can.

Central Illinois

M. M. Gilbert, 175 North St., Peoria, Illinois

Chicago

F. E. Ertsman, Chicago Section, SAE, 1420 Fisher Bldg., 343 S. Dearborn St., Chicago 4, Ill.

Cincinnati

L. W. Thorne, 615 Maple Ave., Cincinnati 29, Ohio

Cleveland

Miss C. M. Hill, 7016 Euclid Ave.,

Turn to Page 116

King, Jr., Robert Sherman, Warren H. Smith.

Foreign

William John Collins, Brazil, S. A.; Henry M. Haan, Belgium; M. Sreenivasa Rao, India.

HAVE YOU Changed Your Address?

So that your SAE mail will reach you with the least possible delay, please keep SAE Headquarters and the Secretary of your local Section or Group advised of any changes in your address. Such notices should be sent to:

1. Society of Automotive Engineers, Inc., 29 West 39th St., New York 18, N. Y.
2. The Secretary or Assistant Secretary of your Section or Group at the addresses listed below:

Baltimore

R. L. Ashley, Ashley Chevrolet Sales, Inc., 2001 N. B'way, Baltimore 10, Md.

British Columbia

John B. Tompkins, British Columbia Section, SAE, 1010 Dominion Building, 207 W. Hastings St., Vancouver, B. C., Canada

Buffalo

C. J. Lane, 1807 Elmwood Ave., Buffalo 7, N. Y.

Canadian

F. G. King, Maclean-Hunter Publishing Co., Ltd., 481 University Ave., Toronto, 2, Ont., Can.

Central Illinois

M. M. Gilbert, 175 North St., Peoria, Illinois

Chicago

F. E. Ertsman, Chicago Section, SAE, 1420 Fisher Bldg., 343 S. Dearborn St., Chicago 4, Ill.

Cincinnati

L. W. Thorne, 615 Maple Ave., Cincinnati 29, Ohio

Cleveland

Miss C. M. Hill, 7016 Euclid Ave.,

Turn to Page 116

Follow the lead of these manufacturers who

Specify FRAM*

Allis-Chalmers Mfg. Co.
American LaFrance Foamite Corp.
B. F. Avery & Sons Company
Brockway Motor Company
Buda Company
Chicago Pneumatic Co.
Clark Equipment Company
Continental Motors Corp.
Cooper-Bessemer Co.
Crosley Motors, Inc.
DeLaval Separator Co.
Detroit Tractor Co.
Diamond T Motor Car Co.
Divco Corp.
Euclid Road Machinery Co.
Fate-Root-Heath Co.
Harry Ferguson, Inc.
Ford Motor Co.
Ford Tractor Division
Four Wheel Drive Co.
Hercules Motors Corp.
Hill Diesel Engine Co.
Hobart Bros. Co.
Frank G. Hough Co.
Hudson Motor Car Co.
Hyster Company
International Diesel Electric Co.
Insley Mfg. Co.
Kermath Mfg. Co.
Landis Tool Co.
Lathrop Engine Works
Leader Tractor Co.
Lincoln Electric Co.
Lincoln-Mercury Div. of Ford
Mack Mfg. Co.
McKay Machine Co.
Meriam Instrument Co.
National Supply Co.
Nordberg Mfg. Co.
Oliver Corp.
D. W. Onan & Sons
Owens Yacht Co.
Reo Motors
Schramm, Inc.
Scripps Motors, Inc.
Seagrave Corp.
R. H. Sheppard Co., Inc.
Star Marine Engine Works
Studebaker Corp.
Tropic-Aire, Inc.
Universal Motors Corp.
U. S. Thermo Control Co.
Waukesha Motor Co.
Willys-Overland Motors
Witte Engine Works

Prominent manufacturers listed at left equip some or all of their vehicles with filters embodying the Fram Principle of Oil Filtration.

Investigate Fram Oil & Motor Cleaners

Protect your engines with this same proven method... Fram Oil Filtration. Let us show you how Fram Oil Filters, engineered for your specific problems, can add life and cut maintenance on the engines you manufacture. Write for complete details and technical data. FRAM CORPORATION, Providence 16, R. I. *In Canada:* J. C. Adams Co., Ltd., Toronto, Ontario.

For Complete Engine Protection

FRAM

OIL - AIR - FUEL

FILTERS

Room 210, Cleveland 3, Ohio

Dayton

F. W. Brooks, 1651 Windsor Rd., Dayton 6, Ohio

Detroit

Mrs. S. J. Duvall, 100 Farnsworth Ave., Detroit 2, Michigan

Hawaii

Charles R. Baptiste, Schuman Carriage Co., Ltd., Beretania & Richards St., P. O. Box 2420, Honolulu, T. H.

Indiana

R. P. Atkinson, 6217 N. Delaware St., Indianapolis 20, Indiana

Kansas City

Donald G. Reed, 1113 Minnesota Ave., Kansas City, Kansas

Metropolitan

F. F. Smith, Society of Automotive Engineers, 29 W. 39th St., N. Y. 18, N. Y.

Mid-Continent

H. C. Baldwin, 609 W. South Ave., Ponca City, Oklahoma

Milwaukee

R. K. McConkey, 715 N. Van Buren St., Milwaukee 2, Wisconsin

Montreal

F. H. Moody, 2471 Mayfair Ave., Montreal 28, Que., Canada

New England

E. G. Moody, Edward G. Moody & Son, Inc., Daniel Webster Highway, Box 130, Nashua, N. H.

Northern California

Donald Wimberly, Calif. Research Corp., 200 Bush St., San Francisco 4, Calif.

Northwest

A. D. McLean, 1621 45th St., S. W., Seattle 6, Washington

Oregon

J. B. Clark, Consolidated Freightways, P. O. Box 3618, Portland 8, Oregon

Philadelphia

R. W. Donahue, Sun Oil Co., Auto. Lab., Marcus Hook, Pa.

Pittsburgh

W. J. Kittredge, Jr., 3701 Liberty Ave., Pittsburgh, Pa.

St. Louis

W. H. Cowdery, 735 Brownell Ave., Glendale 22, Missouri

Salt Lake

D. C. Despain, Holsum Bread Co., 935 Denver St., Salt Lake City 4, Utah

Williamsport

A. E. Sieminski, 342 Eldred St., Williamsport, Pa.

San Diego

Joseph S. Skurky, 1811 Coolidge St., San Diego 11, Calif.

Southern California

F. H. Ott, 428 Woodruff Ave., Arcadia, Calif.

Southern New England

A. M. Watson, 130 Porter St., Manchester, Conn.

Spokane-Intermountain

D. F. Hume, S. 1312 S. E. Boulevard, Spokane 10, Washington

Syracuse

D. T. Doman, Shellmans Dr., R. D. #1, Clay, New York

Texas

R. W. Hoyt, Double Seal Ring Co., P. O. Box 566, Ft. Worth, Texas

Twin City

S. Reed Hedges, 5133 10th Ave., S., Minneapolis 7, Minn.

Virginia

F. M. Hutcheson, Motor Parts Corp., 1839 W. Broad St., Richmond 20, Va.

Washington

J. B. Hulse, Truck-Trailer Mfrs. Assn., Inc., 809 National Press Bldg., Washington 4, D. C.

Western Michigan

W. H. Kennedy, 2709 Pinehurst Rd., Muskegon, Michigan

Wichita

G. W. Jones, 225 S. Glendale, Wichita, Kansas

SAE Groups

Atlanta

Z. T. Layfield, Layfield's Garage, Inc., 141 Marietta St., N. W., Atlanta 3, Ga.

Colorado

F. E. Raglin, Public Serv. Co. of Colorado, 1110 W. Third Ave., Denver, Colo.

Mohawk-Hudson

Frank Baker, 1152 Baker Ave., Schenectady 8, N. Y.

For the Sake of Argument

How to Ask for Information

By Norman G. Shidle

No place does "do-unto-others" technique bring more superior results than in letters asking for information. The easier we make it for people to help us, the more help we get.

Replies to a direct mail sales effort are likely to be inversely proportional to the mental and physical work required to return an order. More will make a check mark than will write in a word. More will order two items if the same letter totals the two prices than if it lists the two prices without a total.

Written requests for information produce best results only when the writer does *his* share of thinking and research. Specific questions always get more specific answers than do general inquiries.

Students often write: "Please send me all the information you have about the automobile industry. I need this in connection with a thesis I am writing." Usually they get little they need as compared to the experienced researcher-by-mail who says:

Can you tell me:

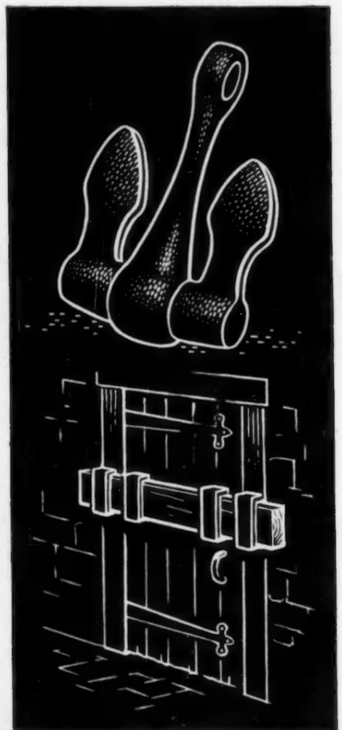
(1) Average compression ratio of engines in 1950 pas-passenger cars;
(2) Average compression ratio of engines in 1940 cars;
(3) And so on, and so on.

One lazy letter-writer even got an irate response from an engineer, amounting to: "If you are not interested enough to figure out what you want to know, you may be sure I'm not. You're not asking for information. You're asking me to write an outline for your thesis and then fill it in."

Adult lazy letter-writers are almost as common—and get the same mediocre results. Almost every engineer has been on the receiving end of letters written by lazy thinkers . . . like the one from compilers of a new encyclopedia who will "appreciate your sending us such source material concerning your field as you may deem appropriate for inclusion in a work of this type."

A letter-reader must know exactly what we want before he can possibly give us a useful, specific answer. If we aren't sure ourselves, we can't expect him to be. . . And if he isn't sure—without too much mental effort—he just doesn't bother to make a real reply. It's no skin off his nose—unless we are customers!

An anchor for paint . . .



a bar against rust . . .

BONDERITE belongs

under the finish of fine painted metal products

The enemies of paint begin to work as soon as your product leaves your plant. Moisture, abrasion, scratches and dents injure the paint film and lay the metal bare. The finish on untreated metal will quickly deteriorate.

On Bonderite-treated metal the paint is anchored by the crystalline phosphate coating, integral with the metal. Because it's non-metallic, Bonderite resists rust and corrosion, confines finish damage from scratches to the injury itself. Bonderite-treated products *look better longer.*

Bonderite is the *standard* corrosion resistant paint base. It's used on thousands of painted metal products. Learn how it can help make *your* product better. Write today.

PROTECTED BY Bonderite PARKER RUST PROOF CO. QUALITY CORROSION RESISTANT

Bonderite, Parco, Parco Lubrite—Reg. U.S. Pat. Off.

PARKER

PARKER RUST PROOF COMPANY
2181 East Milwaukee Ave.
Detroit 11, Michigan

BONDERITE—Corrosion Resistant Paint Base • PARCO COMPOUND—Rust Resistant • PARCO LUBRITE—Wear Resistant for Friction Surfaces